US005810120A

United States Patent [19]

Jamieson et al.

[11] **Patent Number: 5,810,120**

[45] **Date of Patent: Sep. 22, 1998**

[54] **ROLLER GUIDE ASSEMBLY FEATURING A COMBINATION OF A SOLENOID AND AN ELECTROMAGNET FOR PROVIDING COUNTERBALANCED CENTERING CONTROL**

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[73] Assignee: **Otis Elevator Company**, Farmington, Conn.

[21] Appl. No.: **741,751**

[22] Filed: **Nov. 5, 1996**

[51] **Int. Cl.**^{6} **B66B 1/34**; B66B 7/04

[52] **U.S. Cl.** **187/292**; 187/394; 187/410

[58] **Field of Search** 187/409, 410, 187/394, 391, 292

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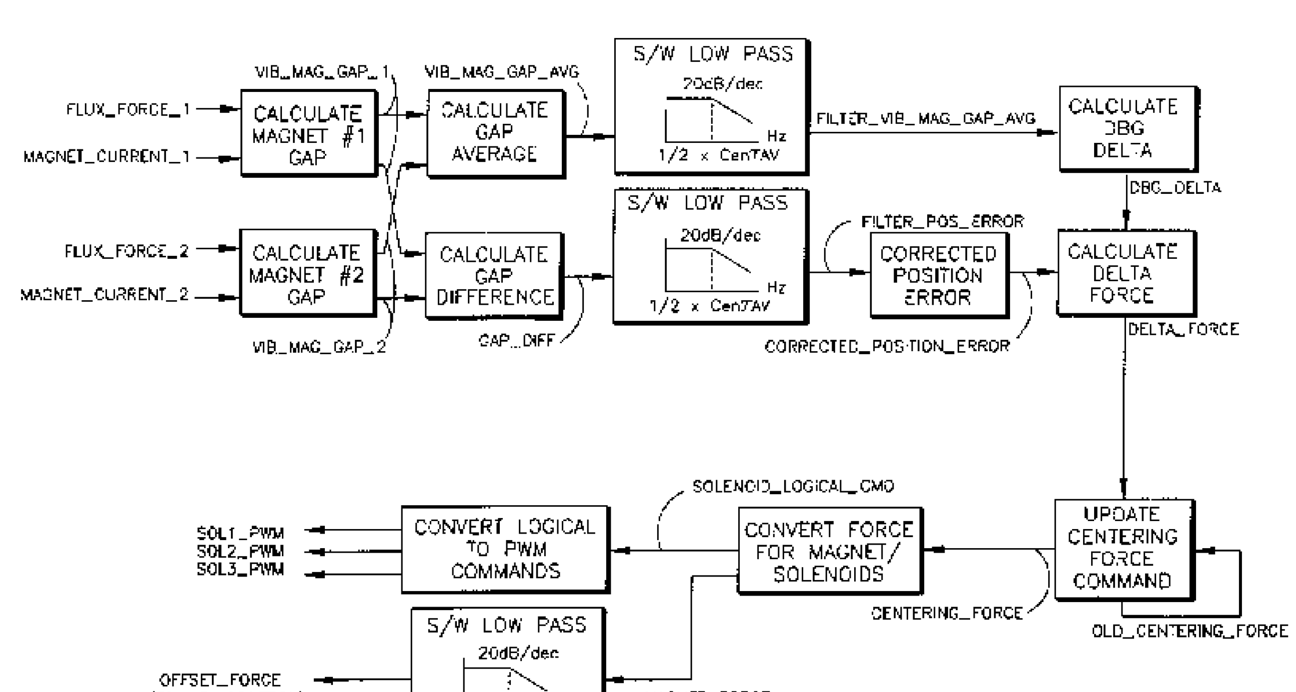
Primary Examiner—Robert Nappi
Attorney, Agent, or Firm—Francis J. Maguire, Jr.

[57] **ABSTRACT**

The present invention provides a roller guide assembly for controlling the position of an elevator car in relation to guide rails of an elevator hoistway in an elevator system. The roller guide assembly includes at least one electromagnet means and at least one solenoid means. The at least one electromagnet means responds to an elevator car control signal from an elevator car controller, for providing at least one electromagnet force to adjust the position of the elevator car in relation to the guide rails of the elevator hoistway. The at least one solenoid means responds to the elevator car control signal, for providing at least one solenoid force that counterbalances said at least one electromagnet force for adjusting the position of the elevator car in relation to the guide rails of the elevator hoistway.

21 Claims, 14 Drawing Sheets

Microfiche Appendix Included
(1 Microfiche, 21 Pages)



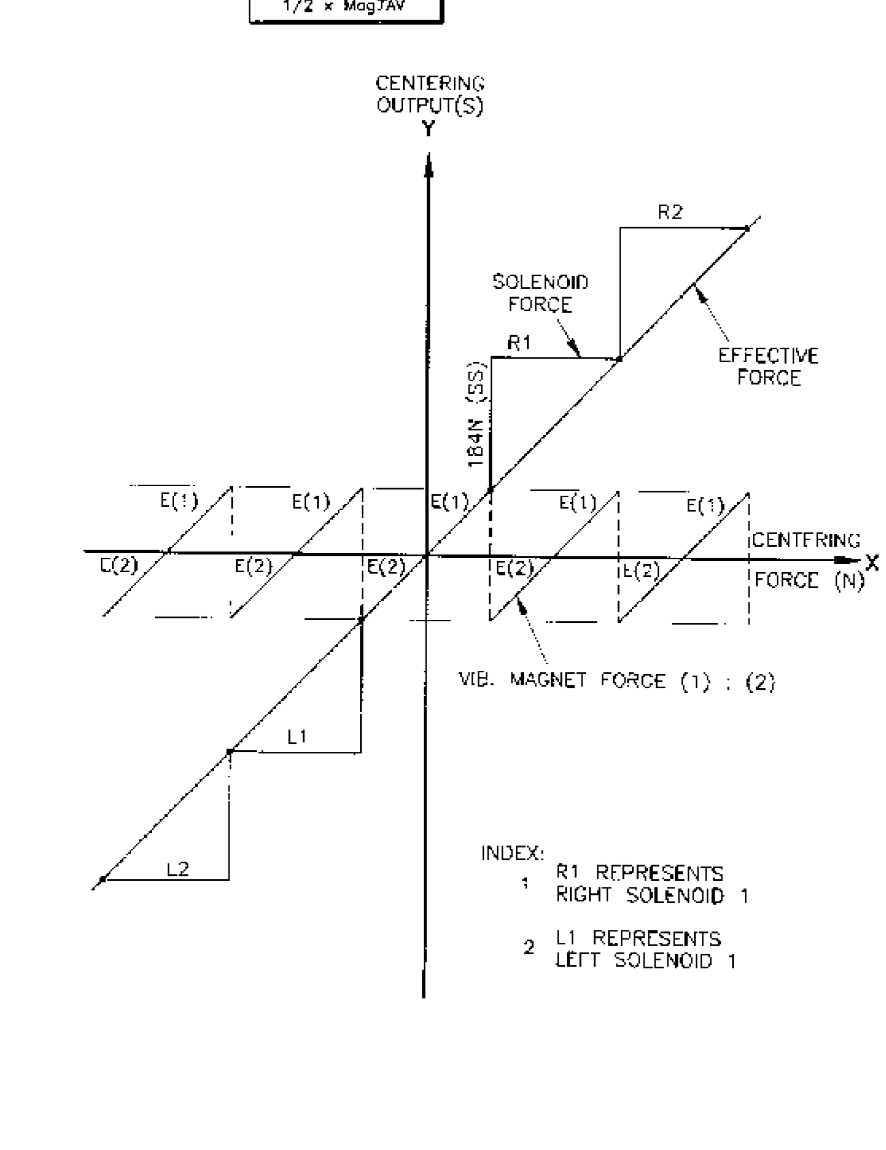
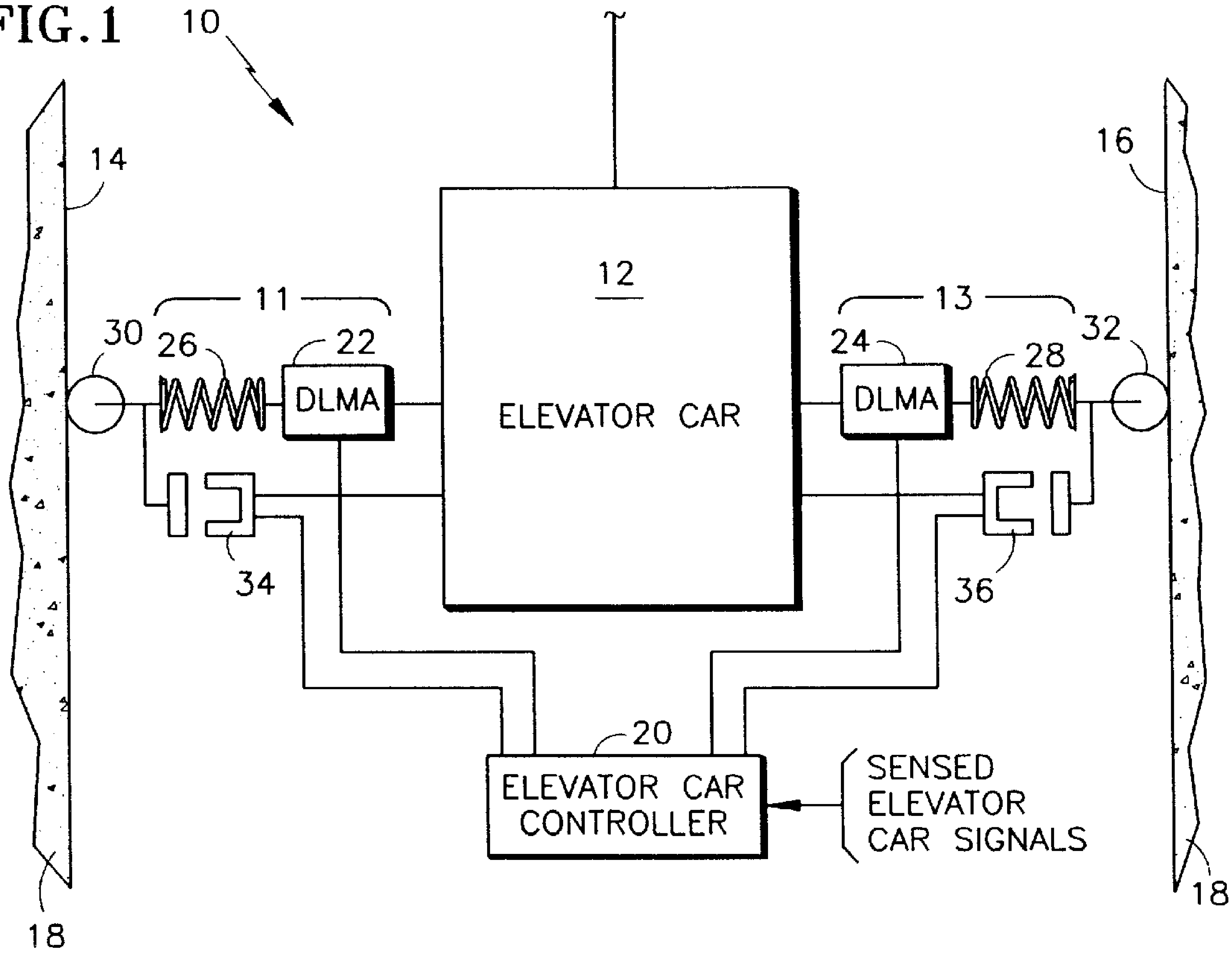
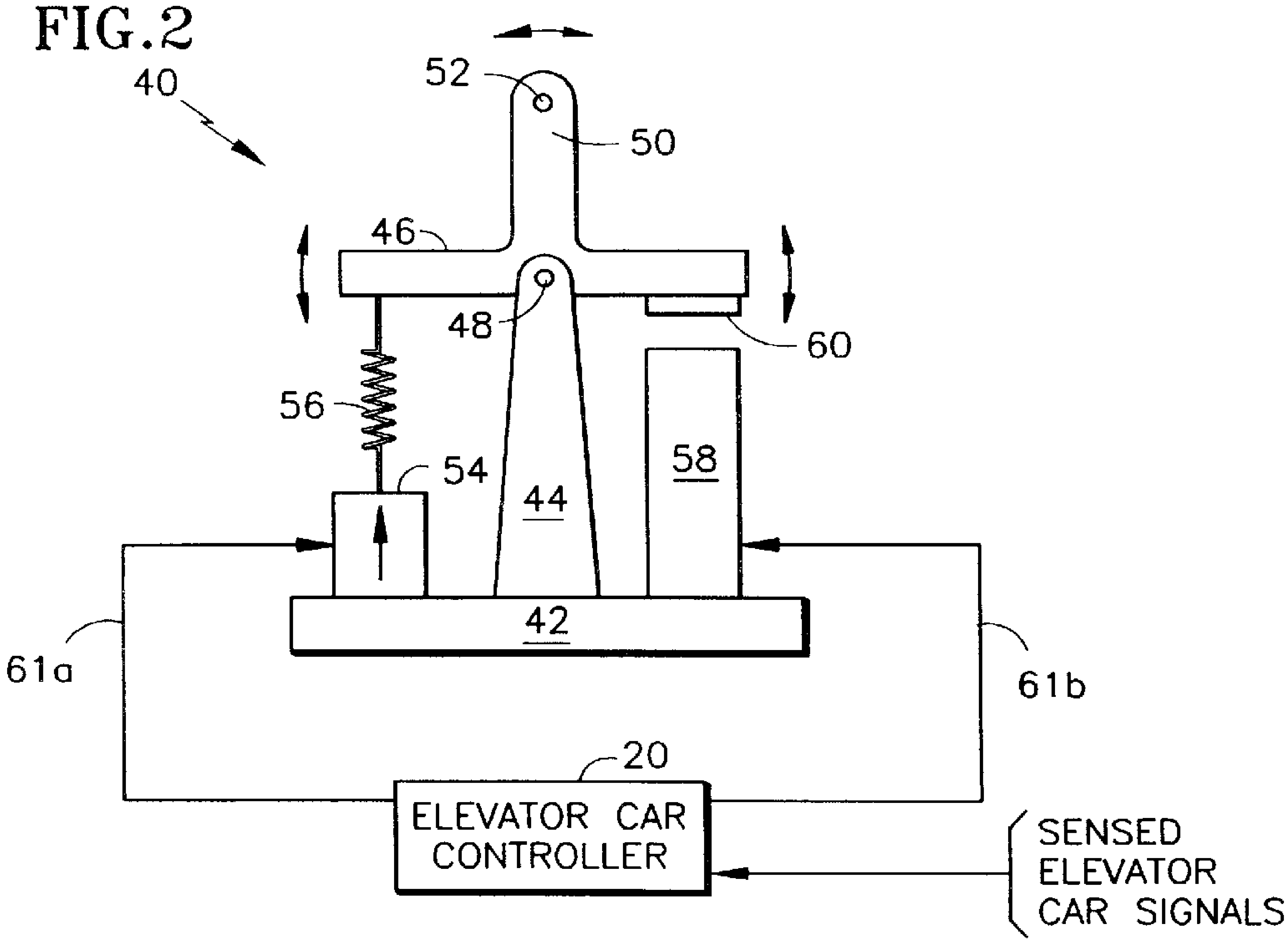


FIG. 1



10
14
16
12
ELEVATOR CAR
11
13
30
26
22
DLMA
24
DLMA
28
32
34
36
20
ELEVATOR CAR CONTROLLER
SENSED ELEVATOR CAR SIGNALS
18
18

FIG. 2



40
52
50
46
48
60
56
54
44
58
42
61a
61b
20
ELEVATOR CAR CONTROLLER
SENSED ELEVATOR CAR SIGNALS

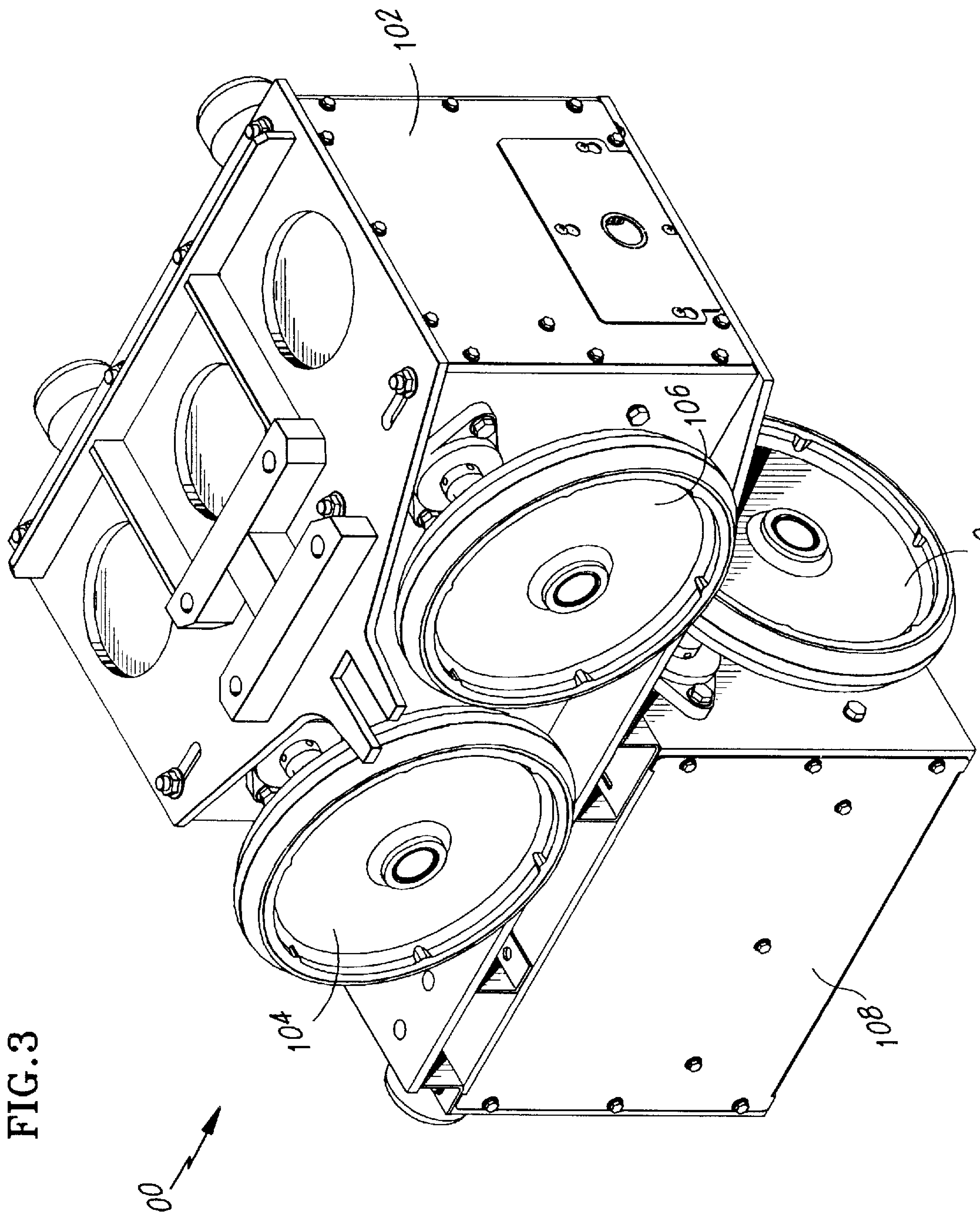
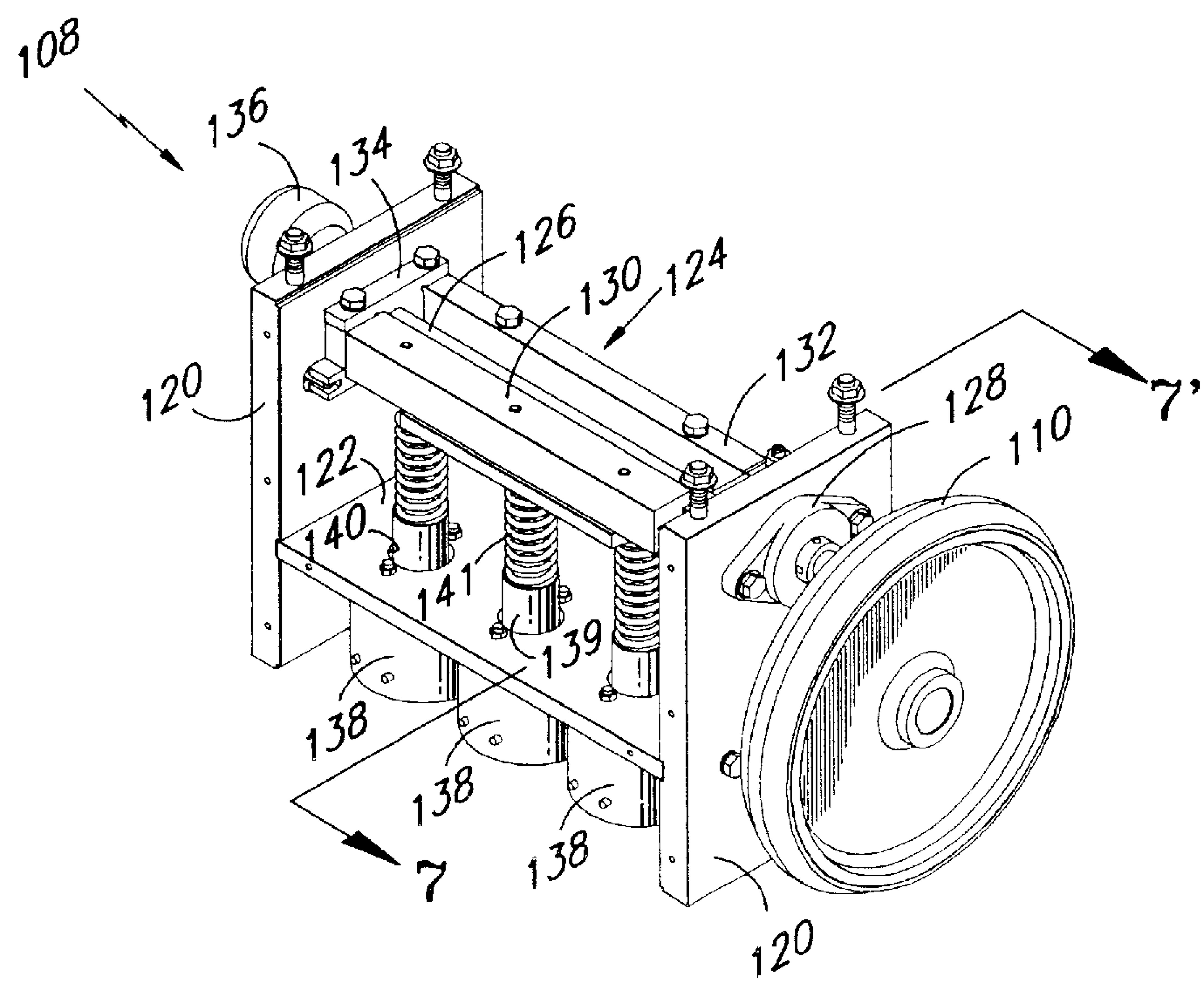


FIG. 3

FIG. 4


108
136
134
126
130
124
132
7'
128
110
120
122
140
141
139
138
138
138
7
120

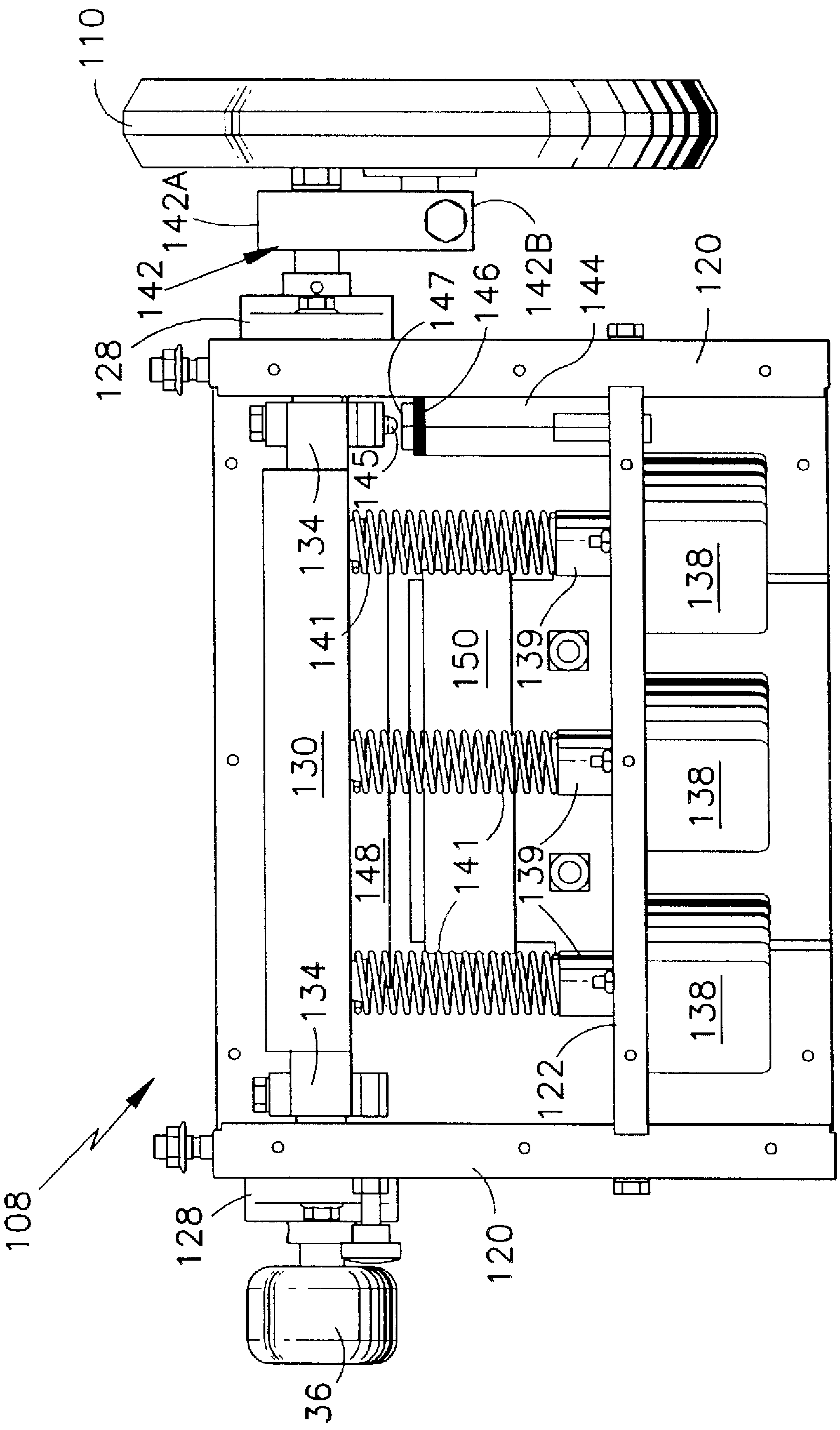

110
142A
142
142B
147
146
144
120
128
145
134
141
150
139
138
130
148
122
108
136

FIG.5

FIG. 6

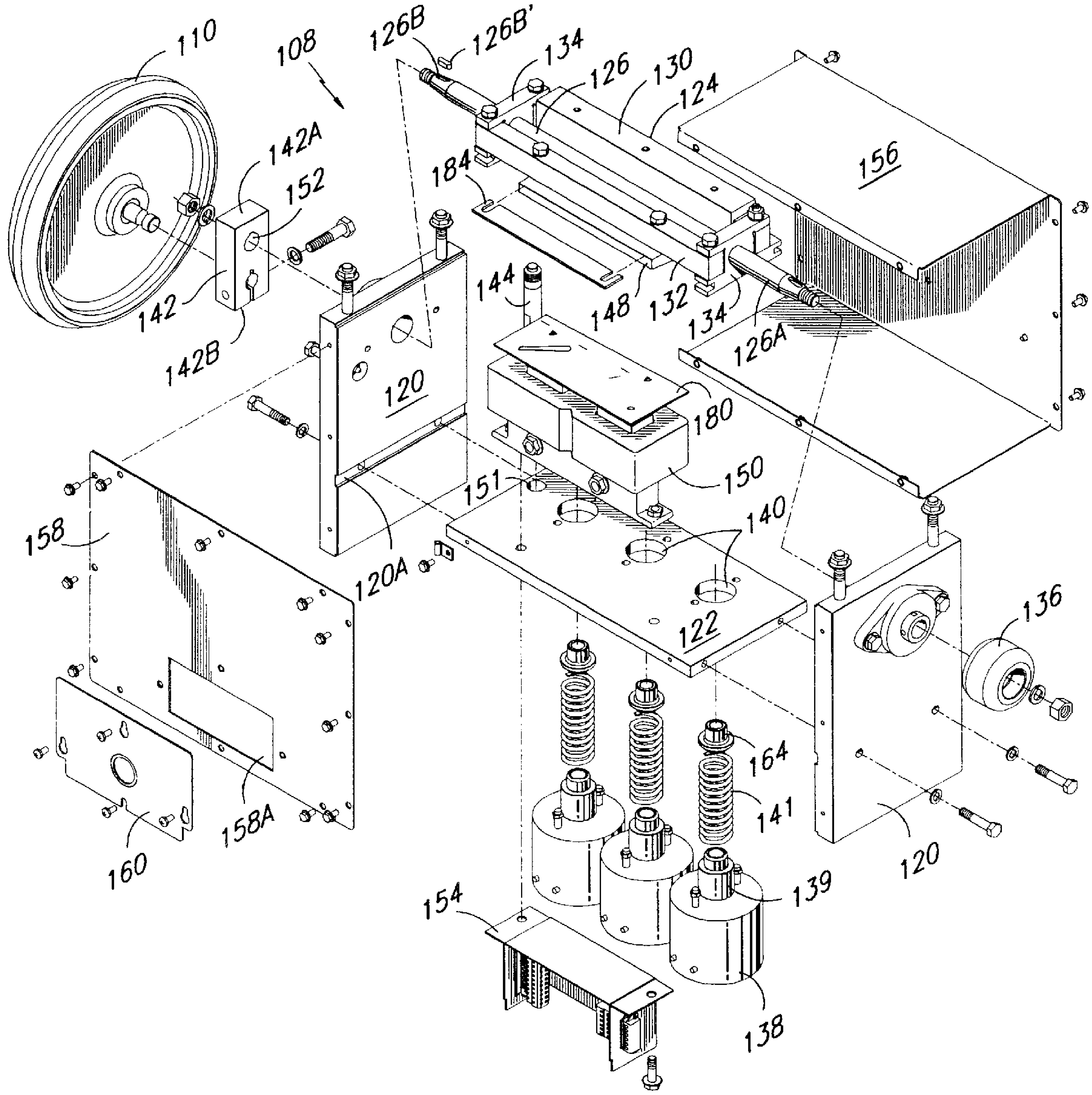

110
108
126B
126B'
134
126
130
124
156
142A
152
184
142
142B
144
148
132
134
126A
120
180
150
151
140
158
120A
122
136
164
141
120
139
160
158A
154
138

FIG.7

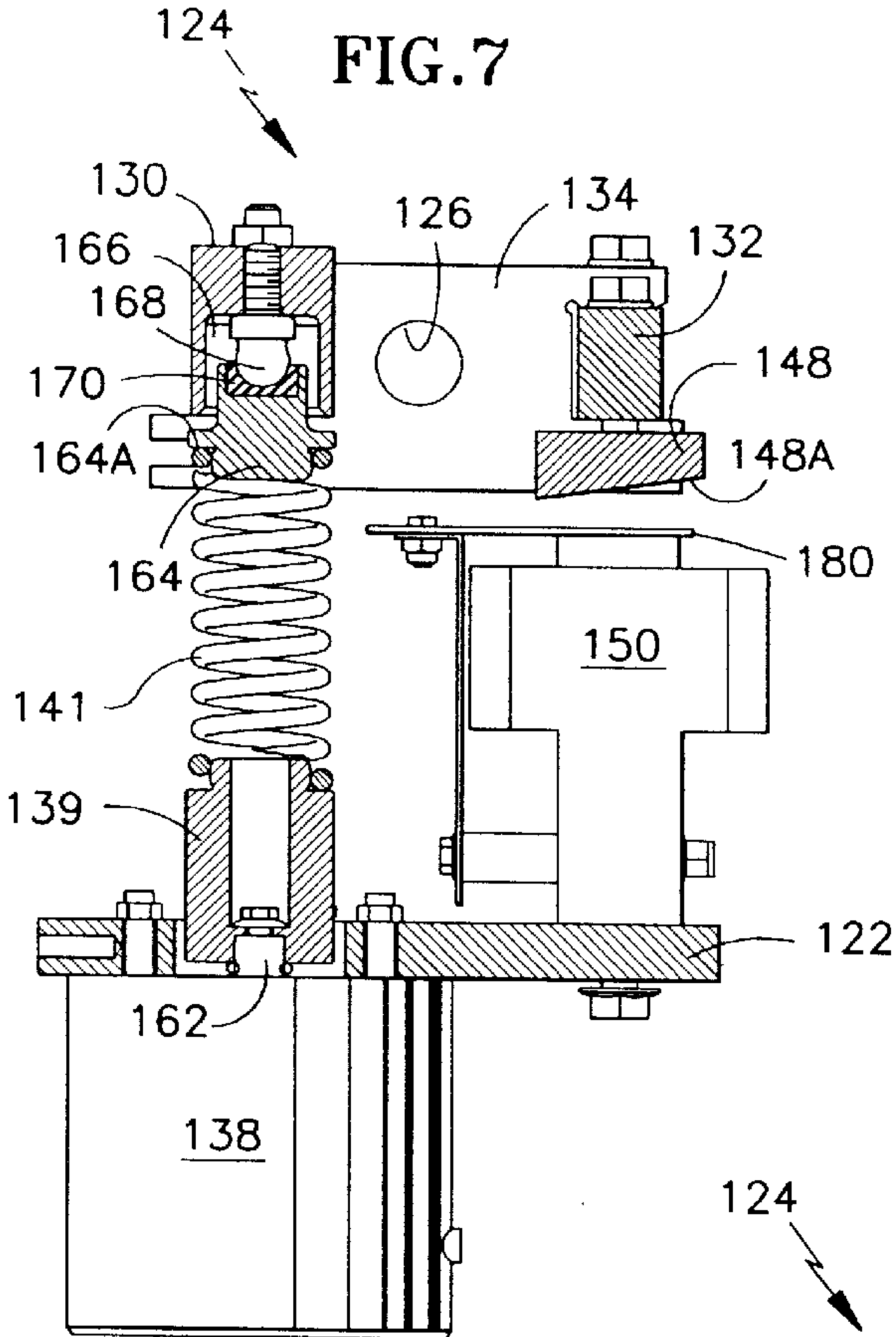


FIG.8

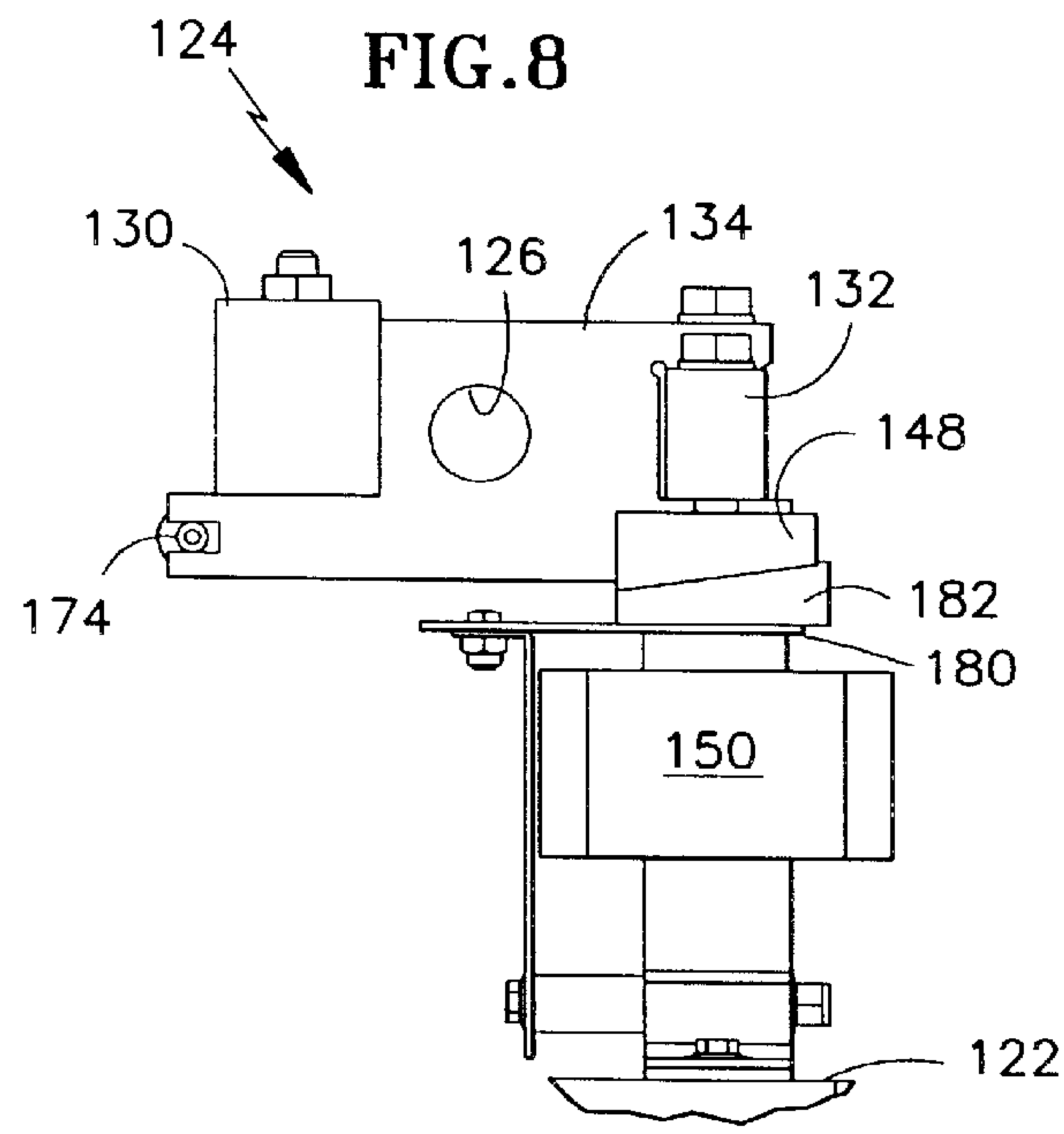


FIG.9

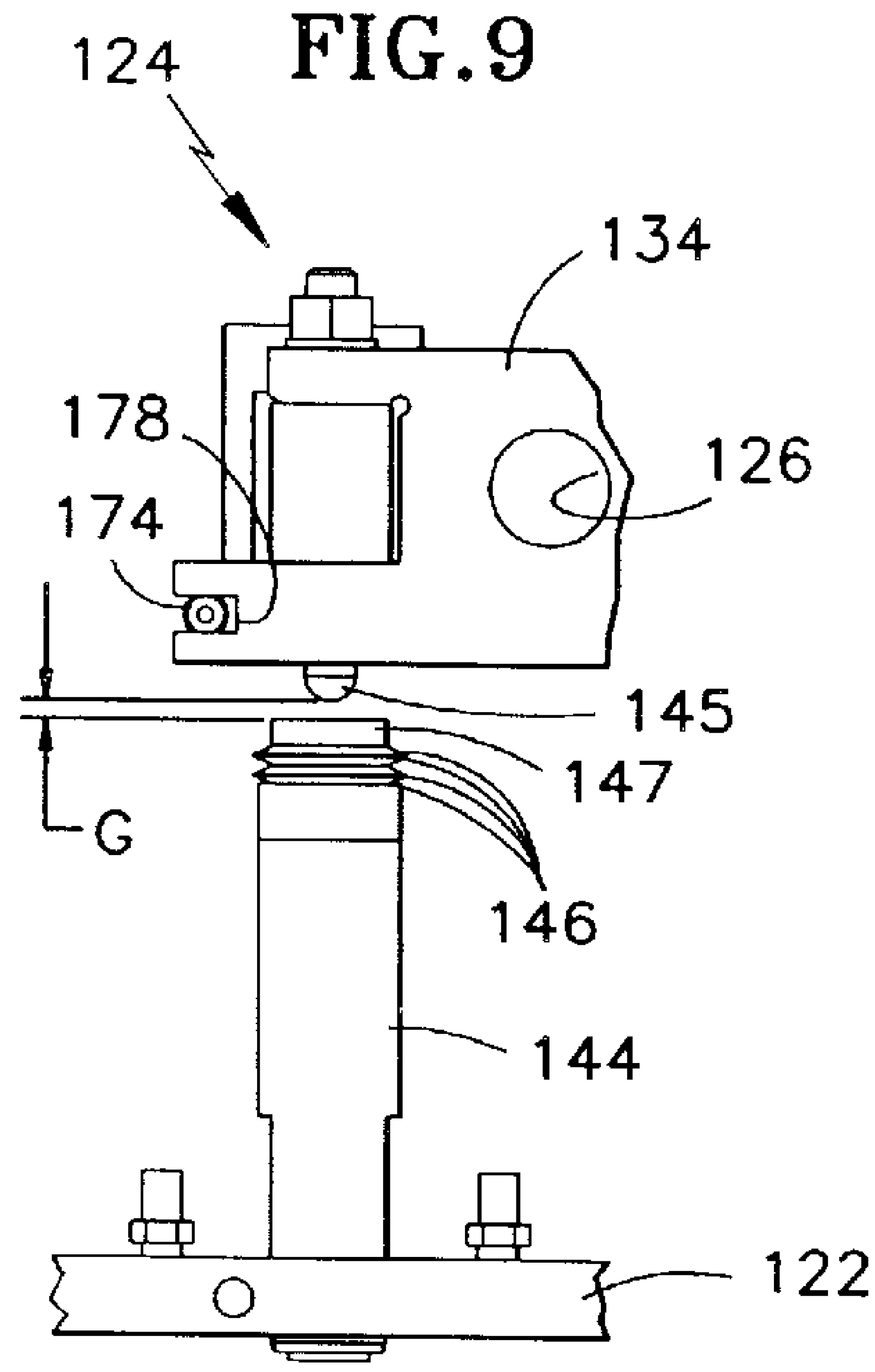


FIG. 10

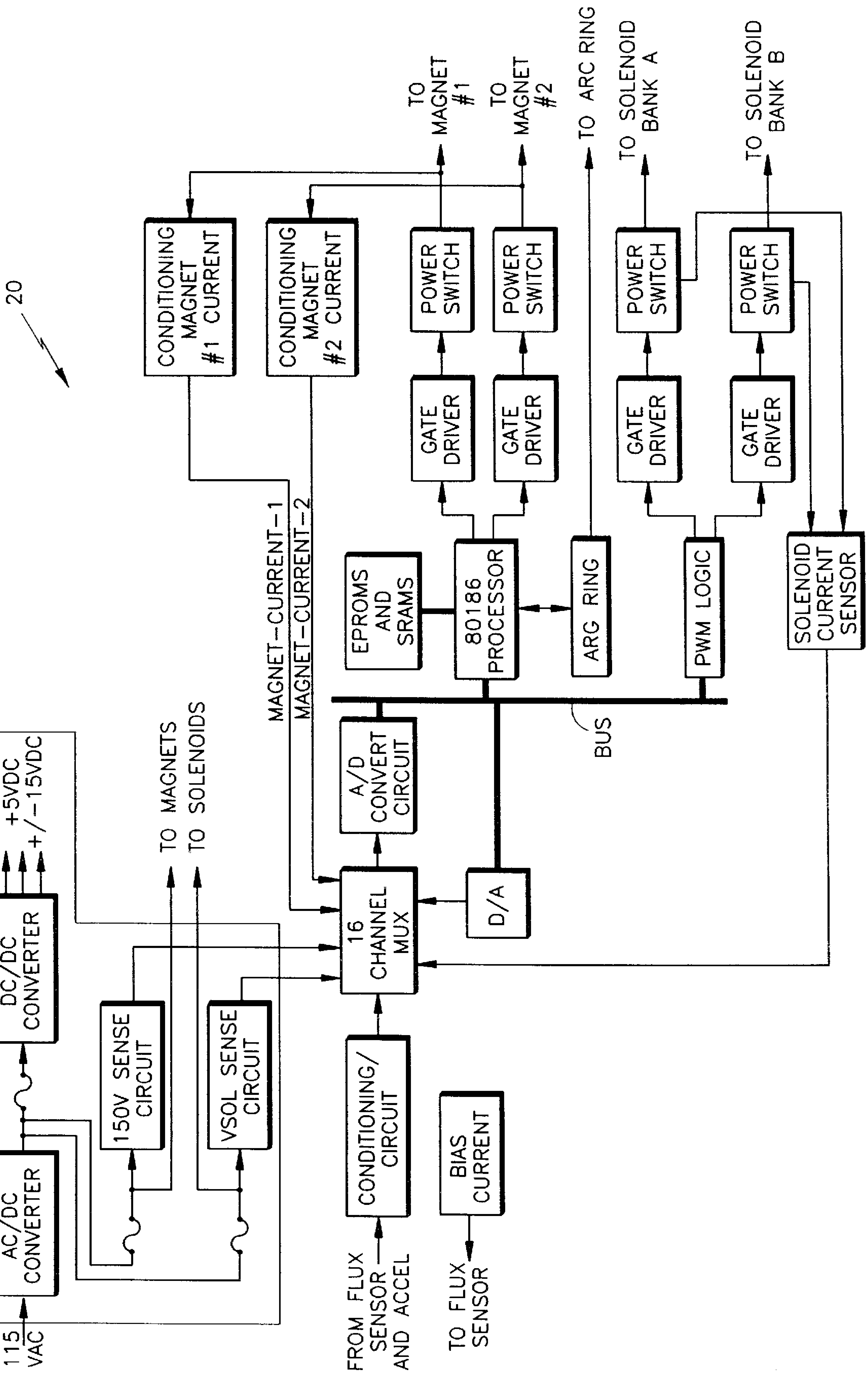

20
CONDITIONING MAGNET #1 CURRENT
CONDITIONING MAGNET #2 CURRENT
POWER SWITCH
GATE DRIVER
TO MAGNET #1
TO MAGNET #2
TO ARC RING
TO SOLENOID BANK A
TO SOLENOID BANK B
EPROMS AND SRAMS
80186 PROCESSOR
ARG RING
PWM LOGIC
SOLENOID CURRENT SENSOR
MAGNET-CURRENT-1
MAGNET-CURRENT-2
BUS
A/D CONVERT CIRCUIT
D/A
16 CHANNEL MUX
+5VDC
+/-15VDC
TO MAGNETS
TO SOLENOIDS
DC/DC CONVERTER
150V SENSE CIRCUIT
VSOL SENSE CIRCUIT
AC/DC CONVERTER
115 VAC
CONDITIONING/ CIRCUIT
BIAS CURRENT
FROM FLUX SENSOR AND ACCEL
TO FLUX SENSOR

FIG. 11

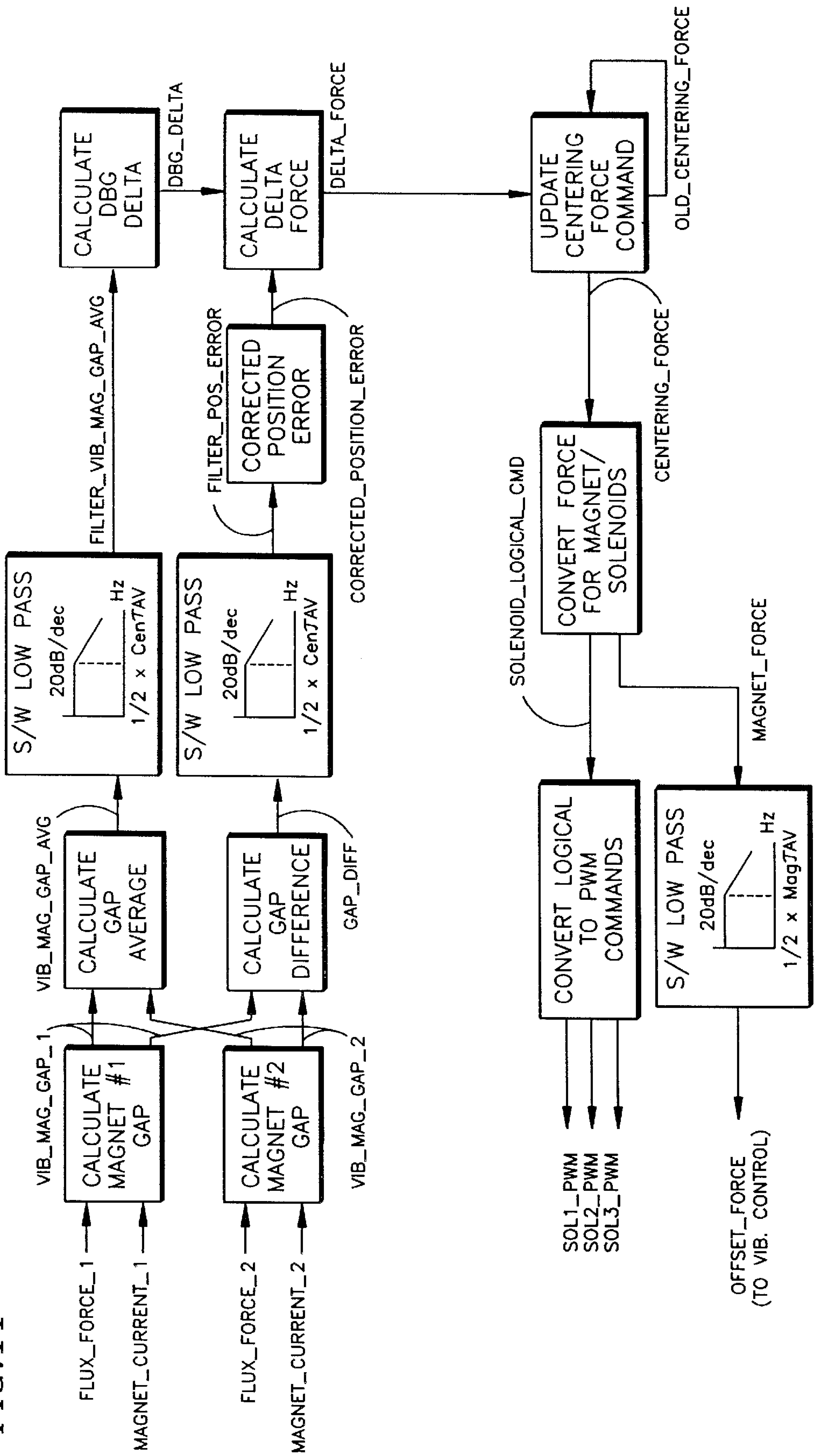
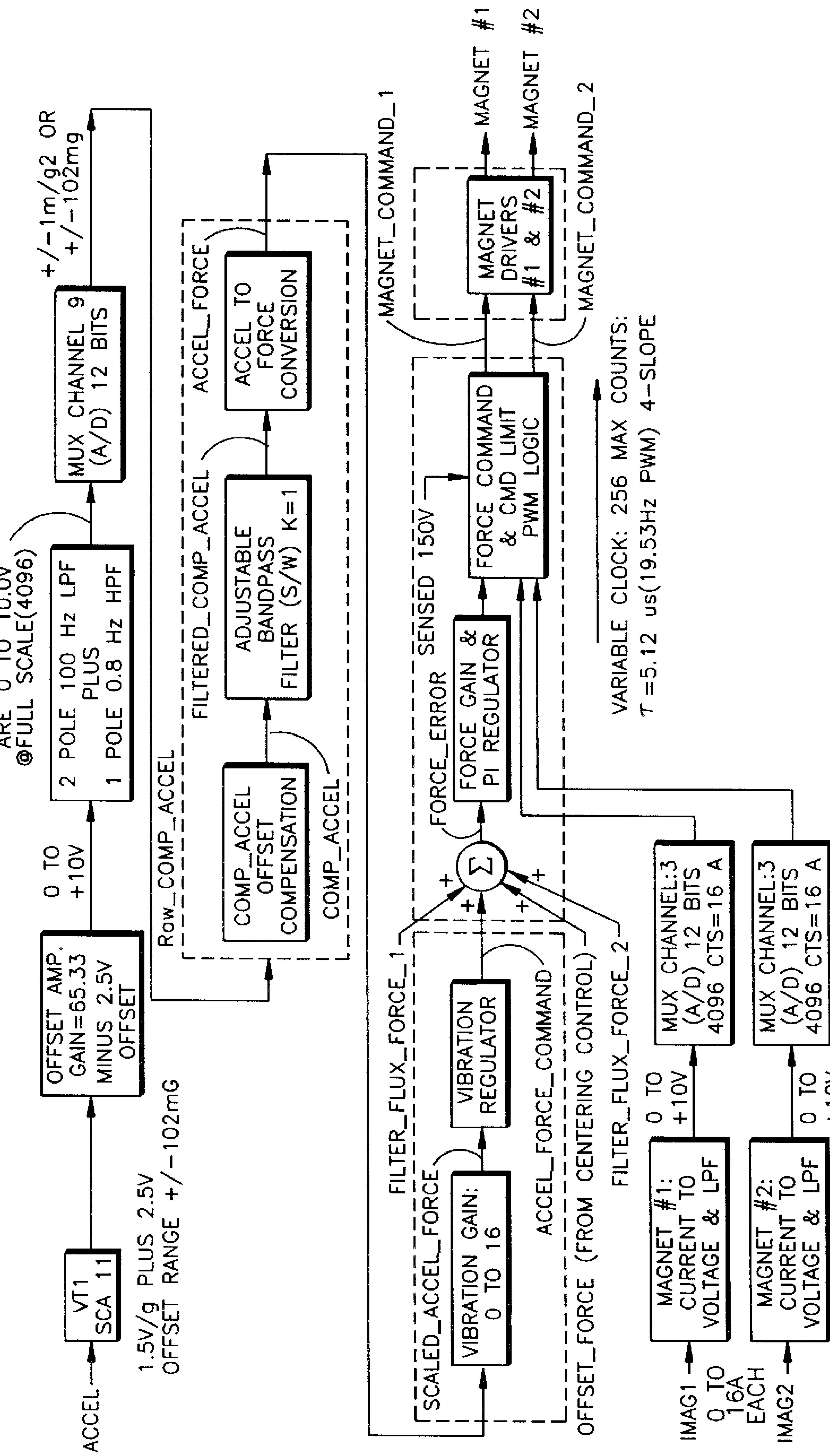

FLUX_FORCE_1
MAGNET_CURRENT_1
FLUX_FORCE_2
MAGNET_CURRENT_2
CALCULATE MAGNET #1 GAP
CALCULATE MAGNET #2 GAP
VIB_MAG_GAP_1
VIB_MAG_GAP_2
CALCULATE GAP AVERAGE
CALCULATE GAP DIFFERENCE
VIB_MAG_GAP_AVG
GAP_DIFF
S/W LOW PASS
20dB/dec
Hz
1/2 x CenTAV
S/W LOW PASS
20dB/dec
Hz
1/2 x CenTAV
FILTER_VIB_MAG_GAP_AVG
FILTER_POS_ERROR
CORRECTED POSITION ERROR
CORRECTED_POSITION_ERROR
CALCULATE DBG DELTA
DBG_DELTA
CALCULATE DELTA FORCE
DELTA_FORCE
UPDATE CENTERING FORCE COMMAND
OLD_CENTERING_FORCE
CENTERING_FORCE
CONVERT FORCE FOR MAGNET/ SOLENOIDS
SOLENOID_LOGICAL_CMD
MAGNET_FORCE
CONVERT LOGICAL TO PWM COMMANDS
SOL1_PWM
SOL2_PWM
SOL3_PWM
S/W LOW PASS
20dB/dec
Hz
1/2 x MagTAV
OFFSET_FORCE
(TO VIB. CONTROL)

FIG.12A


ACCEL
VT1 SCA 11
1.5V/g PLUS 2.5V OFFSET RANGE +/-102mG
OFFSET AMP. GAIN=65.33 MINUS 2.5V OFFSET
0 TO +10V
2 POLE 100 Hz LPF PLUS 1 POLE 0.8 Hz HPF
AU A/D CHANNNELS ARE 0 TO 10.0V @FULL SCALE(4096)
MUX CHANNEL 9 (A/D) 12 BITS
+/-1m/g2 OR +/-102mg
Raw_COMP_ACCEL
COMP_ACCEL OFFSET COMPENSATION
COMP_ACCEL
ADJUSTABLE BANDPASS FILTER (S/W) K=1
FILTERED_COMP_ACCEL
ACCEL TO FORCE CONVERSION
ACCEL_FORCE
SCALED_ACCEL_FORCE
VIBRATION GAIN: 0 TO 16
VIBRATION REGULATOR
ACCEL_FORCE_COMMAND
OFFSET_FORCE (FROM CENTERING CONTROL)
FILTER_FLUX_FORCE_1
FILTER_FLUX_FORCE_2
+
+
+
+
Σ
FORCE_ERROR
FORCE GAIN & PI REGULATOR
SENSED 150V
FORCE COMMAND & CMD LIMIT PWM LOGIC
MAGNET_COMMAND_1
MAGNET_COMMAND_2
MAGNET DRIVERS #1 & #2
MAGNET #1
MAGNET #2
VARIABLE CLOCK: 256 MAX COUNTS:
τ =5.12 us(19.53Hz PWM) 4-SLOPE
IMAG1
0 TO 16A EACH
IMAG2
MAGNET #1: CURRENT TO VOLTAGE & LPF
MAGNET #2: CURRENT TO VOLTAGE & LPF
0 TO +10V
0 TO +10V
MUX CHANNEL:3 (A/D) 12 BITS 4096 CTS=16 A
MUX CHANNEL:3 (A/D) 12 BITS 4096 CTS=16 A

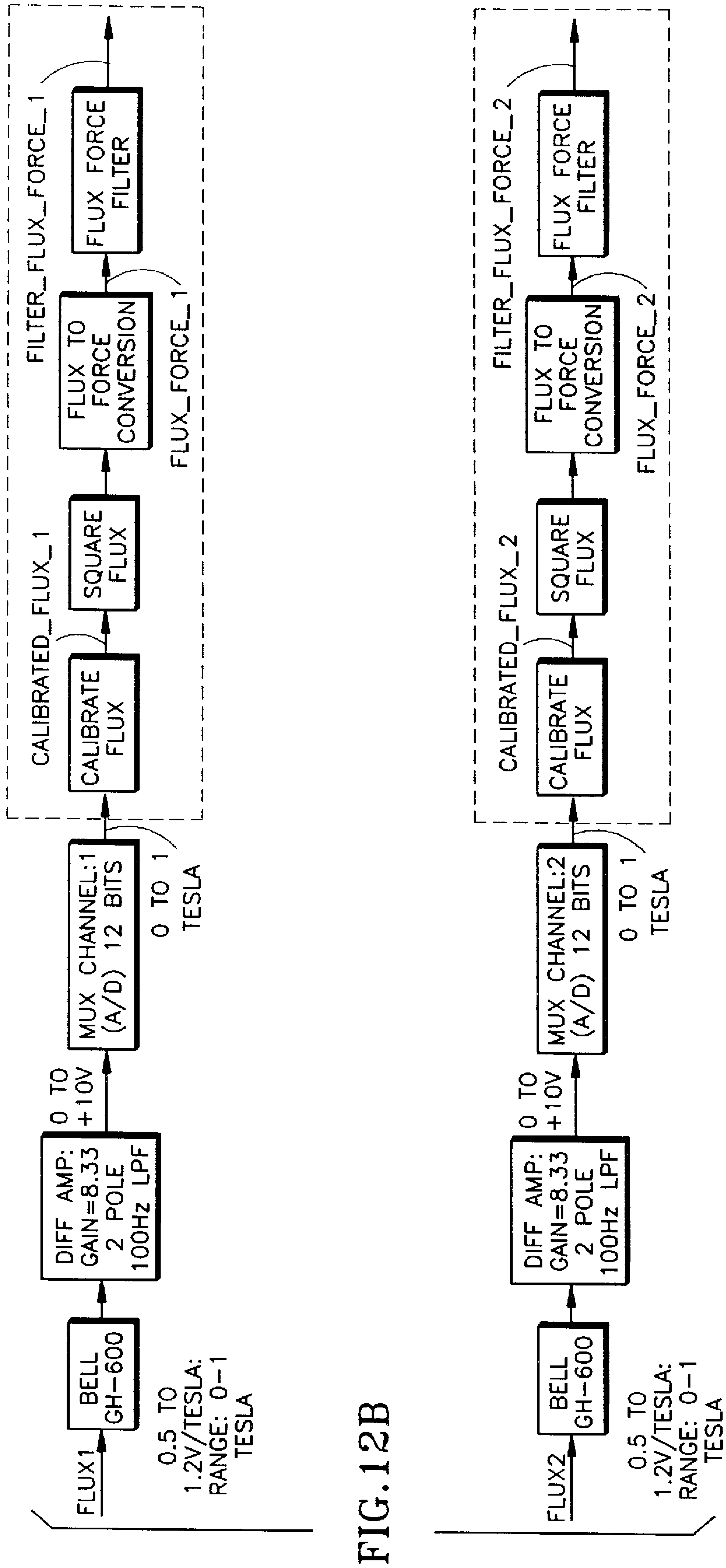

FLUX1
BELL GH-600
0.5 TO 1.2V/TESLA: RANGE: 0-1 TESLA
DIFF AMP: GAIN=8.33 2 POLE 100Hz LPF
0 TO +10V
MUX CHANNEL:1 (A/D) 12 BITS
0 TO 1 TESLA
CALIBRATE FLUX
CALIBRATED_FLUX_1
SQUARE FLUX
FLUX TO FORCE CONVERSION
FLUX_FORCE_1
FLUX FORCE FILTER
FILTER_FLUX_FORCE_1
FLUX2
BELL GH-600
0.5 TO 1.2V/TESLA: RANGE: 0-1 TESLA
DIFF AMP: GAIN=8.33 2 POLE 100Hz LPF
0 TO +10V
MUX CHANNEL:2 (A/D) 12 BITS
0 TO 1 TESLA
CALIBRATE FLUX
CALIBRATED_FLUX_2
SQUARE FLUX
FLUX TO FORCE CONVERSION
FLUX_FORCE_2
FLUX FORCE FILTER
FILTER_FLUX_FORCE_2

FIG.12B

FIG.13

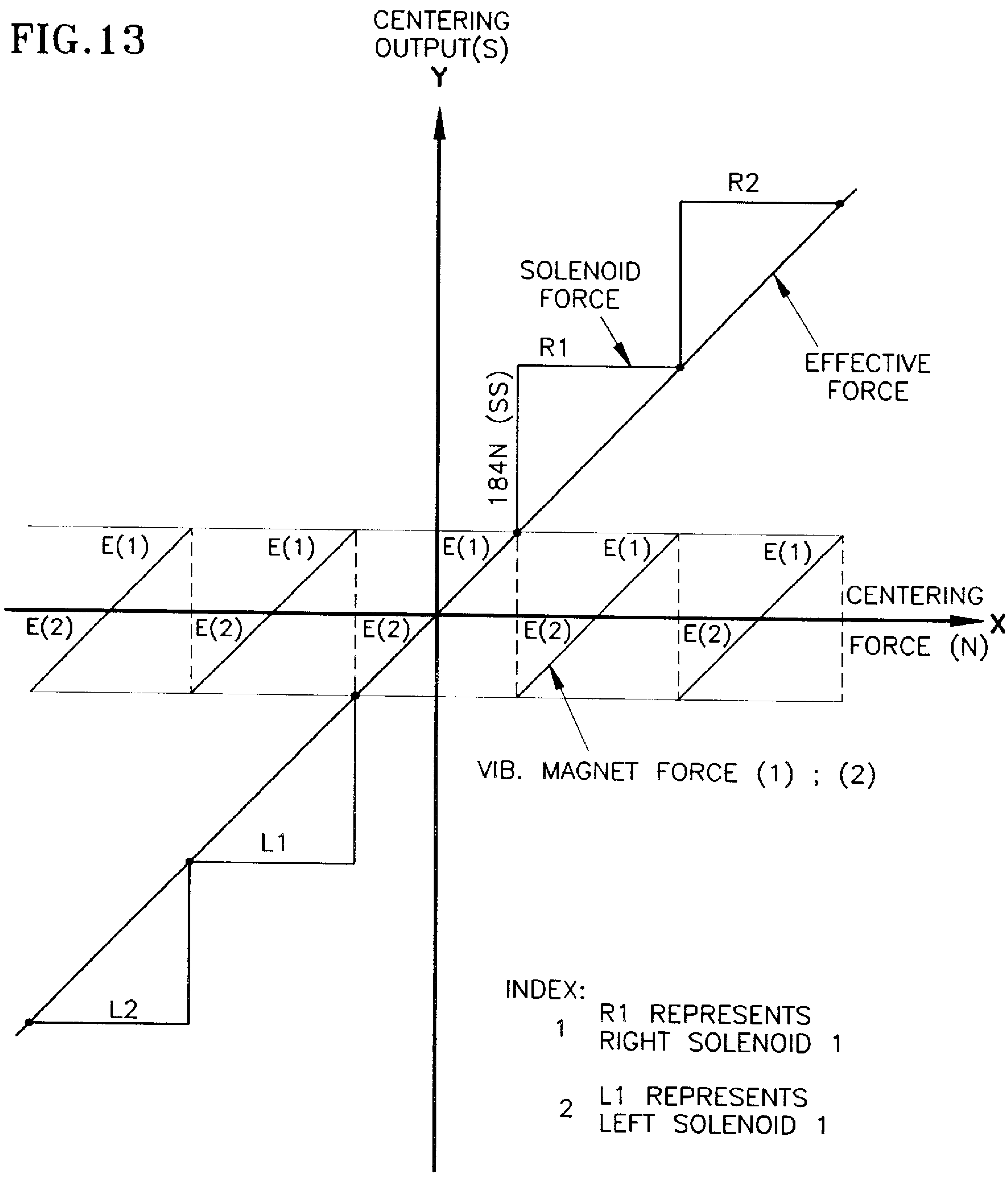

CENTERING
OUTPUT(S)
Y
R2
SOLENOID
FORCE
R1
184N (SS)
EFFECTIVE
FORCE
E(1)
E(1)
E(1)
E(1)
E(1)
E(2)
E(2)
E(2)
E(2)
E(2)
CENTERING
FORCE (N)
X
VIB. MAGNET FORCE (1) ; (2)
L1
L2
INDEX:
1 R1 REPRESENTS
RIGHT SOLENOID 1
2 L1 REPRESENTS
LEFT SOLENOID 1

FIG. 14

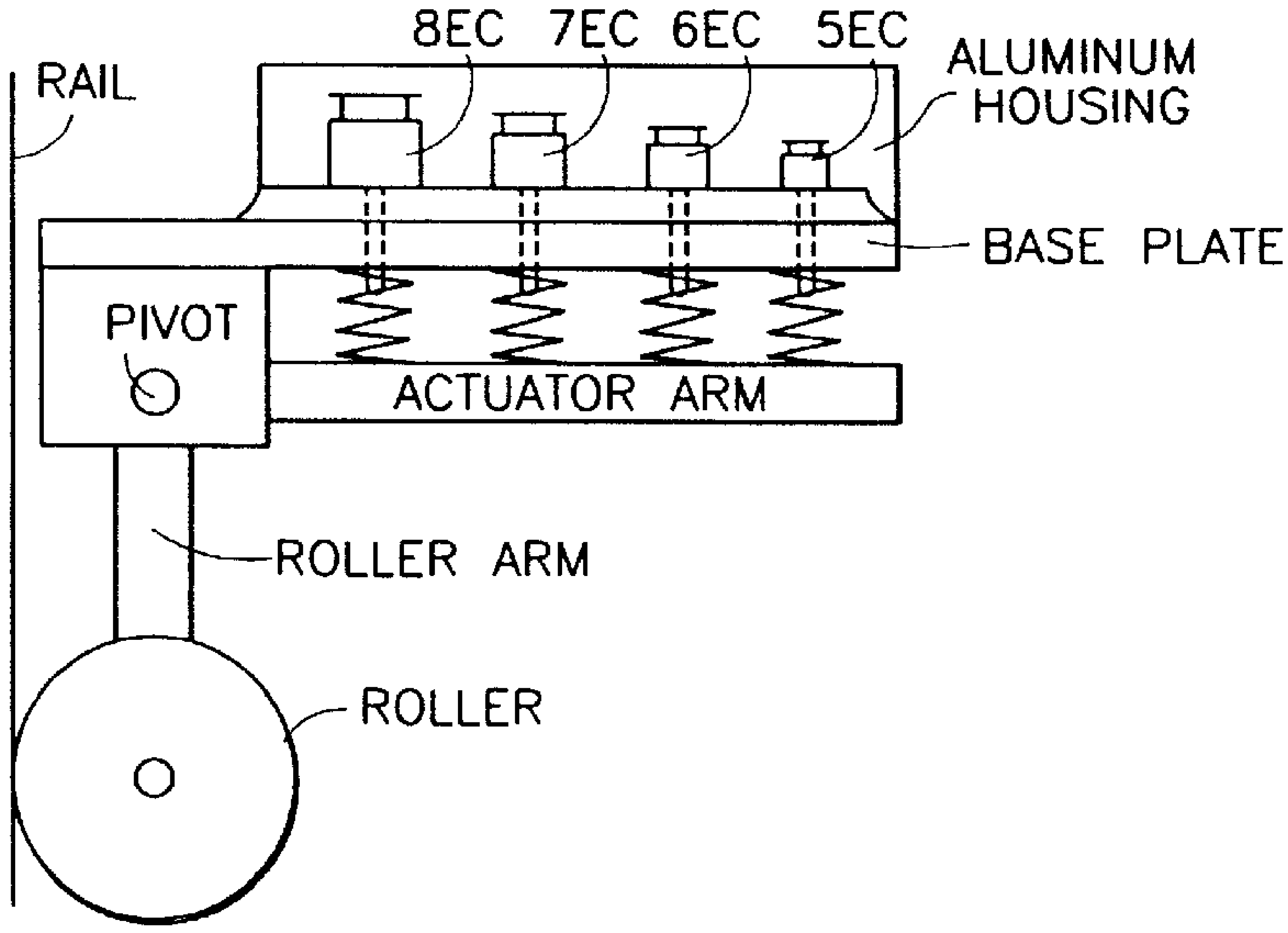

8EC
7EC
6EC
5EC
ALUMINUM HOUSING
RAIL
BASE PLATE
PIVOT
ACTUATOR ARM
ROLLER ARM
ROLLER

FIG. 15

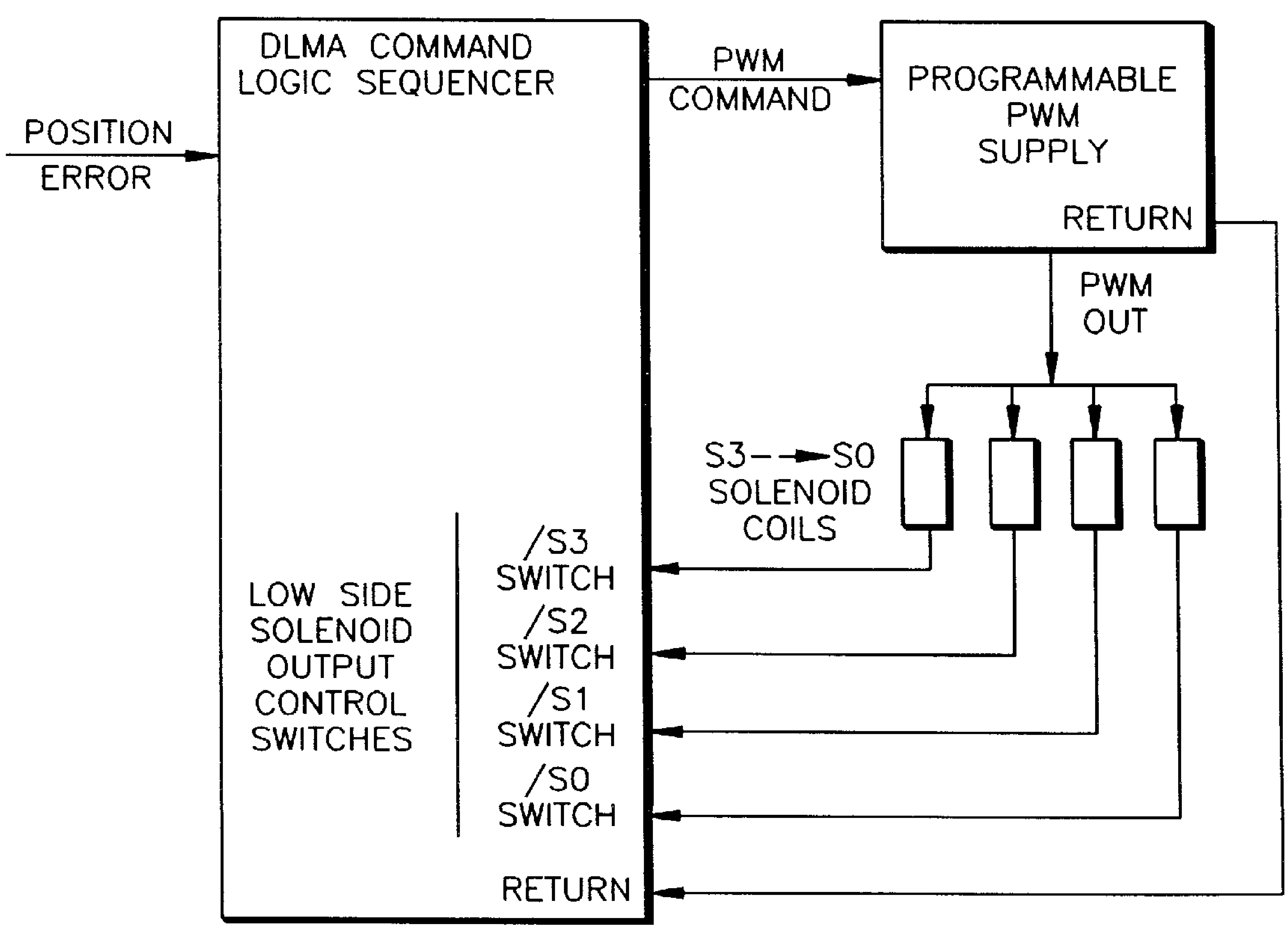

DLMA COMMAND LOGIC SEQUENCER
PWM COMMAND
PROGRAMMABLE PWM SUPPLY
RETURN
POSITION ERROR
PWM OUT
S3--►S0 SOLENOID COILS
LOW SIDE SOLENOID OUTPUT CONTROL SWITCHES
/S3 SWITCH
/S2 SWITCH
/S1 SWITCH
/S0 SWITCH
RETURN

FIG. 16

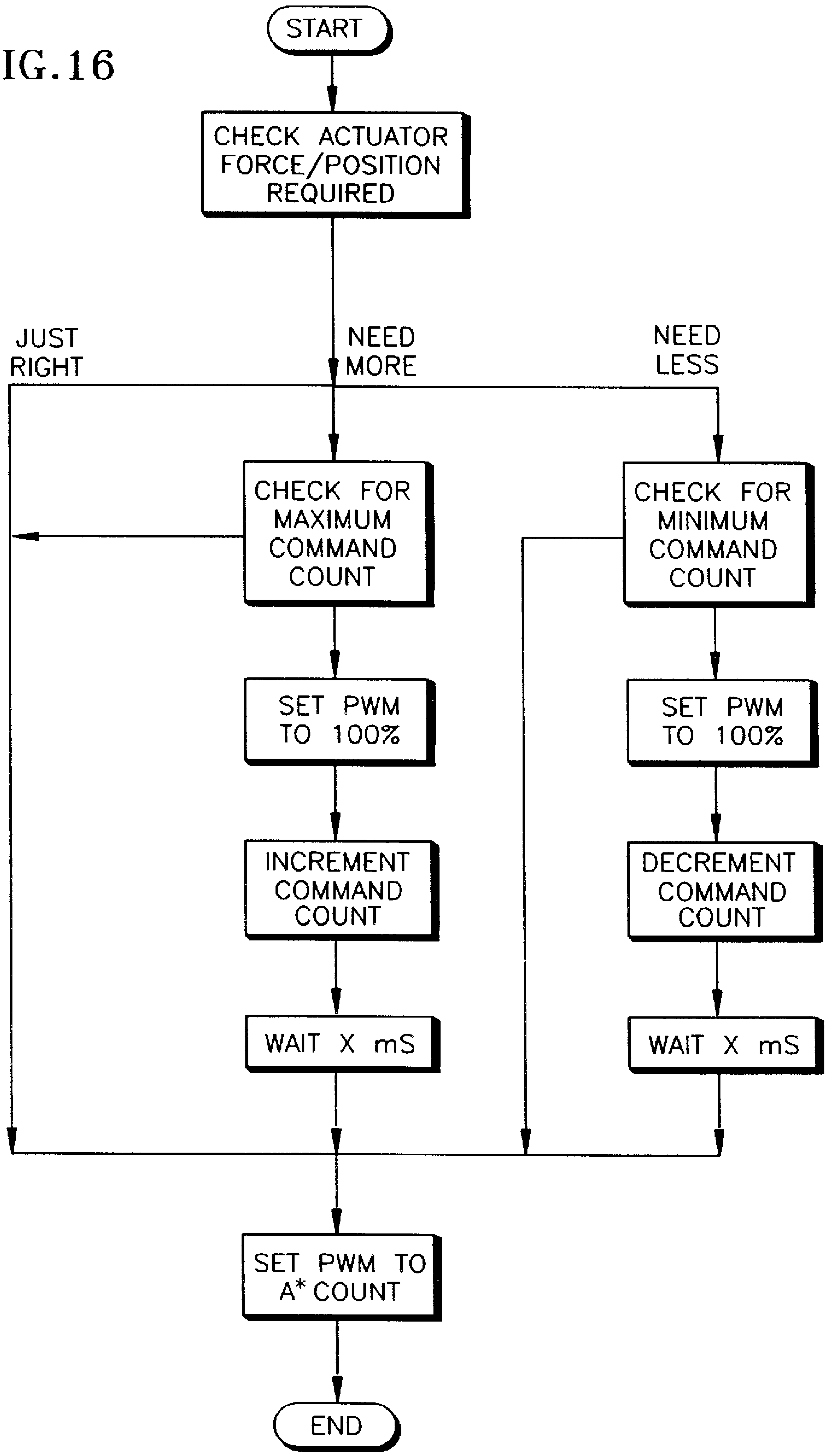

START
CHECK ACTUATOR FORCE/POSITION REQUIRED
JUST RIGHT
NEED MORE
NEED LESS
CHECK FOR MAXIMUM COMMAND COUNT
SET PWM TO 100%
INCREMENT COMMAND COUNT
WAIT X mS
CHECK FOR MINIMUM COMMAND COUNT
SET PWM TO 100%
DECREMENT COMMAND COUNT
WAIT X mS
SET PWM TO A* COUNT
END

FIG. 17

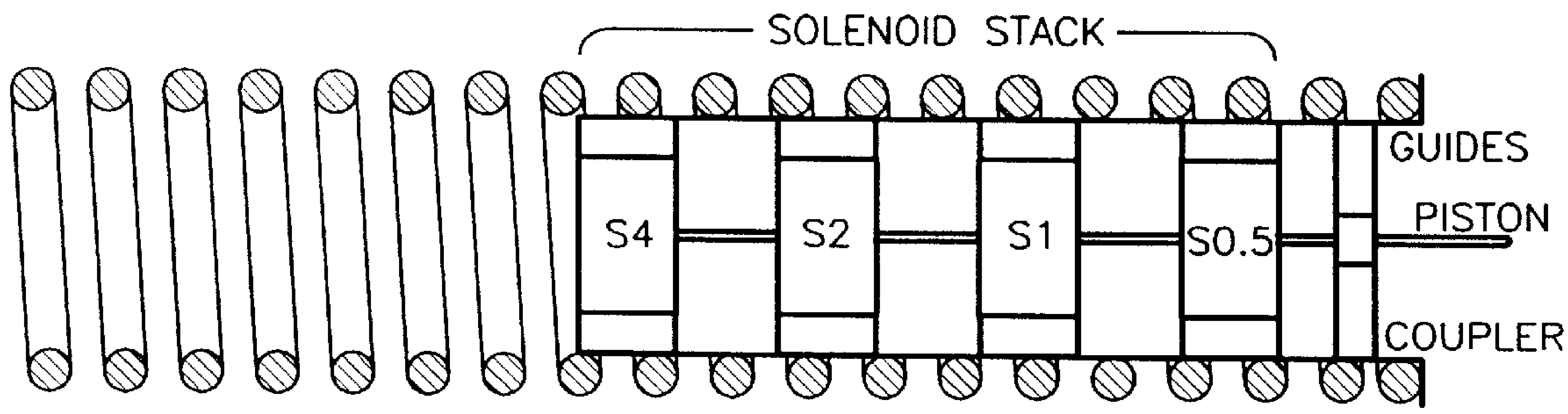
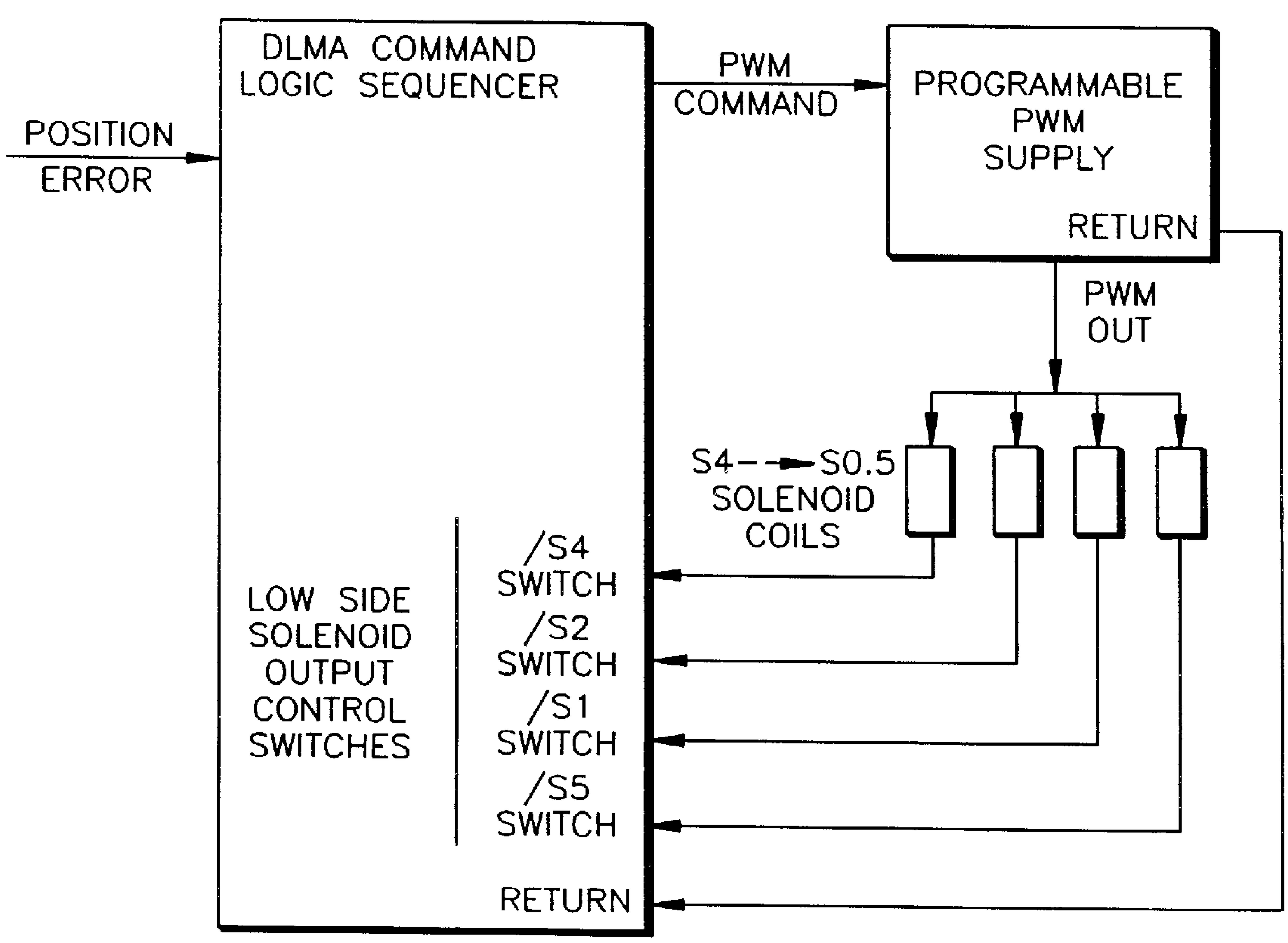

SOLENOID STACK
GUIDES
PISTON
COUPLER
S4
S2
S1
S0.5

FIG. 18


DLMA COMMAND LOGIC SEQUENCER
POSITION ERROR
PWM COMMAND
PROGRAMMABLE PWM SUPPLY
RETURN
PWM OUT
S4→S0.5 SOLENOID COILS
LOW SIDE SOLENOID OUTPUT CONTROL SWITCHES
/S4 SWITCH
/S2 SWITCH
/S1 SWITCH
/S5 SWITCH
RETURN

ROLLER GUIDE ASSEMBLY FEATURING A COMBINATION OF A SOLENOID AND AN ELECTROMAGNET FOR PROVIDING COUNTERBALANCED CENTERING CONTROL

MICROFICHE APPENDIX

This application contains a Microfiche Appendix having a program listing with 21 pages on one Microfiche.

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BACKGROUND OF THE INVENTION

1. Field of the Invention

The present invention relates generally to an elevator control system, and more particularly to an elevator control system for controlling the position of an elevator car in relation to guide rails of an elevator hoistway.

2. Description of the Prior Art

Elevator systems for controlling the position of an elevator car in relation to guide rails of an elevator hoistway are known in the art. For example, U.S. Pat. No. 5,117,946 shows and describes an elevator system for an active roller guide having a telescoping ball screw adjustment device (also known as a worm gear) for providing centering control. (See U.S. Pat. No. 5,117,946, FIG. 5, reference numeral 52.) The telescoping ball screw adjustment device is controlled by an electric motor and adjusts the force of a coil spring applied on a roller guide for centering the elevator car in relation to the guide rails of the elevator hoistway. The subject matter of U.S. Pat. No. 5,117,946 is hereby incorporated by reference.

However, the telescoping ball screw adjustment device in U.S. Pat. No. 5,117,946 has some disadvantages, such as high ratio gear boxes, brushless DC motors, brakes, clutches as well as complex control circuitry, which is very expensive in terms of weight, cost and reliability. For instance, the high ratio gears in the telescoping ball screw adjustment actuator device and the brushless DC motor for driving the telescoping ball screw adjustment actuator device may require regular time-consuming and expensive maintenance which is further exacerbated when an active roller guide wheel assembly is located underneath the elevator car. The brushless DC motor also requires complex control circuitry. The high ratio gear boxes and brushless DC motor may also be unreliable over long-term continuous use.

DISCLOSURE OF THE INVENTION

The present invention provides a unique roller guide assembly for an elevator system for controlling the position of an elevator car in relation to guide rails of an elevator hoistway.

In its broadest sense, the roller guide assembly includes at least one electromagnet means and at least one solenoid means. Then at least one electromagnet means responds to an elevator car control signal from an elevator car controller, for providing at least one electromagnet force to adjust the position of the elevator car in relation to the guide rails of the elevator hoistway. Then at least one solenoid means responds to the elevator car control signal, for providing at least one solenoid force that counterbalances said at least one electromagnet force for adjusting the position of the elevator car in relation to the guide rails of the elevator hoistway. In effect, at least one electromagnet means provides a fine adjustment of the position of the elevator car in relation to the guide rails of the elevator hoistway, and the at least one solenoid force provides a coarse adjustment of the position of the elevator car in relation to the guide rails of the elevator hoistway.

In one embodiment, the roller guide assembly also includes at least one coil spring that responds to at least one solenoid force, for providing at least one coil spring force for adjusting the position of the elevator car in relation to the guide rails of the elevator hoistway. The sensed elevator car signal may include information about the position and/or acceleration of the elevator car in relation to the guide rails of the elevator hoistway. The control of the position of the elevator car in relation to the guide rails of the elevator hoistway includes either side-to-side centering control, front-to-back centering control, or both. Then at least one electromagnet also responds to the elevator car control signal, for providing then at least one electromagnet force to control the vibration between the roller (also known as guide wheel) of the elevator car and a guide rail of the elevator hoistway during the control of the position of the elevator car in relation to the guide rails of the elevator hoistway.

One important advantage of using a solenoid for centering control is that it requires significantly less time-consuming and expensive maintenance, in effect having a much longer service life span. The solenoid does not require complex circuitry and is also inherently much more reliable over long continuous use.

Other advantages will become apparent to those skilled in the art from the following detailed description read in conjunction with the appended claims and drawings attached hereto.

BRIEF DESCRIPTION OF THE DRAWINGS

The drawings, not drawn to scale, include:

FIG. **1** is a simplified block diagram of an actuator in an elevator guidance system for an active roller guide application.

FIG. **2** is a simplified actuator diagram.

FIG. **3** is a perspective view of an elevator car roller guide assembly cluster **100** of the present invention.

FIG. **4** is a perspective view of a side-to-side roller guide assembly **108** shown in FIG. **3**.

FIG. **5** is a side view of the side-to-side roller guide assembly **108** shown in FIG. **3**.

FIG. **6** is an exploded view of the side-to-side roller guide assembly **108** shown in FIGS. **3–5**.

FIG. **7** is a partial cross sectional view of the side-to-side roller guide assembly **108** shown in FIG. **4** along lines **7–7'**.

FIG. **8** is a view of the electromagnet gap calibration shown in FIG. **7**.

FIG. **9** is a view of the hard stop gap calibration shown in FIG. **5**.

FIG. **10** shows a block diagram of an active roller guide control board for providing centering and vibration control for the elevator car **12** in FIG. **1**.

FIG. **11** is a block diagram of a software algorithm for providing centering control of the elevator **12** in FIG. **1**.

FIG. **12(A)** shows a vibration control overview which may be used in an elevator car controller **20** in FIGS. **1–2**.

FIG. **12(B)** shows a flux sensing overview which may be used to provide filter__flux force signals **1** and **2** in FIG. **10**.

FIG. **13** is a graph depicting centering output force(s) versus a required centering force.

FIG. **14** shows an embodiment of the present invention having parallel solenoid digital linear magnetic actuators (DLMA).

FIG. **15** shows a parallel DLMA control technique for the embodiment shown in FIG. **14**.

FIG. **16** shows a flowchart for a solenoid logical command determination for the embodiment shown in FIG. **14**.

FIG. **17** shows an embodiment of the present invention having a stacked series of solenoid digital linear magnetic actuators.

FIG. **18** shows a series DLMA control technique for the embodiment shown in FIG. **17**.

BEST MODE FOR CARRYING OUT THE INVENTION

FIG. **1** shows an elevator system generally indicated as **10** for controlling the horizontal position of an elevator car **12** in relation to guide rails **14**, **16** of an elevator hoistway **18**.

As shown, the elevator system **10** includes an elevator car controller **20** and active roller guide assemblies **11**, **13**. Each active roller guide assembly **11**, **13** has solenoid assemblies **22**, **24**. The elevator car controller **20** responds to sensed elevator car signals with information about the position of the elevator car **12** in relation to the guide rails **14**, **16** of the elevator hoistway **18**, and provides elevator car control signals with information about the control of the position of the elevator car **12** in relation to the guide rails **14**, **16** of the elevator hoistway **18**. The solenoid assemblies **22**, **24** respond to the elevator car control signals, by providing solenoid forces for adjusting the position of the elevator car **12** in relation to the guide rails **14**, **16** of the elevator hoistway **18**. In the elevator system, the control of the horizontal position of the elevator car **12** in relation to the guide rails **14**, **16** of the elevator hoistway **18** includes either side-to-side centering control, front-to-back centering control, or both.

Each roller guide assembly **11**, **13** also includes linear coil springs **26**, **28** that respond respectively to the solenoid forces, for providing linear coil spring forces for adjusting the horizontal position of the car **12** in relation to the guide rails **14**, **16** of the elevator hoistway **18**.

Each roller guide assembly **11**, **13** also includes electromagnets **34**, **36** that respond to the elevator car control signal, for providing electromagnet forces to control the position of the elevator car **12** via rollers **30**, **32** in relation to the guide rails **14**, **16** of the elevator hoistway **18**, in effect providing a dual control function, as discussed in detail below with respect to FIGS. **10–13**. In effect, the electromagnets **34**, **36** provide a fine position control and the solenoid assemblies **22**, **24** provide a coarse position control. Moreover, the electromagnets **34**, **36** may also respond to the elevator car control signal, by providing electromagnet forces to control the vibration of the car transmitted to the car, e.g. by the rollers **30**, **32** riding on respective guide rails **14**, **16** of the elevator hoistway **18** and due to the bumps in the rail joints.

The sensed elevator car signals shown in FIG. **1** are provided from a typical elevator car sensing means which is known in the art, including that shown and described in U.S.

Pat. No. 5,117,946, hereby incorporated by reference. The sensed elevator car signals can include information about the position and/or acceleration of the elevator car in relation to the guide rails of the elevator hoistway. In the prior art, usually the sensed position signal is used in a low frequency car centering control loop for controlling a relatively high-force but slow acting actuator while the sensed acceleration signal is used in a higher frequency control loop for controlling a relatively low-force but fast-acting actuator, such as an electromagnet.

The elevator car controller **20** for providing the side-to-side centering control, the front-to-back centering control, or both, as well as vibration control is shown and described below with respect to FIGS. **10–13**. In general, the side-to-side centering control, the front-to-back centering control, or both, adjusts the position of the elevator car **12** in relation to the guide rails **14**, **16** of the elevator hoistway **18**, and also with the help of the electromagnet, as described below. The vibration control, also by means of the electromagnet, controls the vibration of the car transmitted by each roller **30**, **32** of the elevator car **12** chiefly due to rail irregularities in a respective guide rail **14**, **16** of the elevator hoistway **18** at the same time as assisting in the control of the centering of the elevator car **12** in relation to the guide rails **14**, **16** of the elevator hoistway **18**. However, the scope of the invention is not intended to be limited to any particular elevator car controller for providing the side-to-side centering control, the front-to-back centering control, or both, as well as for providing vibration control.

FIG. **2** shows a simplified diagram of a roller guide assembly generally indicated as **40** for the purpose of discussing the basic operation of each roller guide assembly **11**, **13** of FIG. **1**. As shown, the actuator assembly **40** includes a solenoid and magnet base **42** having a pivot support **44** mounted thereon. The actuator assembly **40** also includes a pivot bar **46** mounted on the pivot support **44** about a pivot axis **48**, and a roller arm **50** mounted on the pivot bar **46** with a roller axis **52**. The actuator assembly **40** also includes a solenoid **54**, a spring **56**, an electromagnet **58** and a ferromagnetic plate **60**. The actuator assembly **40** is responsive to control signals from the elevator car controller **20** that is in turn responsive to sensed elevator car signals. The control signals include an elevator car controller signal on a line **61*a*** to the solenoid **54** to control the coarse position of the elevator car **12** in relation to the guide rails **14**, **16** of the elevator hoistway **18**, and an elevator car controller signal on a line **61*b*** to the electromagnet **58** to control the vibration and/or the fine position of the elevator car **12** via rollers **30**, **32** in relation to the guide rails **14**, **16** of the elevator hoistway **18**, consistent with that described above with respect to FIG. **1**.

FIG. **3** shows an embodiment of an elevator car guidance assembly cluster of the present invention, generally indicated by the reference numeral **100**. The elevator car guidance assembly cluster **100** includes two front-to-back active roller guide assemblies generally indicated as **102** for controlling front-to-back rollers **104**, **106** with respect to one of the guide rails **14**, **16**. The elevator car guidance assembly cluster **100** also includes a side-to-side active roller guide assembly generally indicated as **108** for controlling a side-to-side roller **110** with respect to one of the guide rails **14**, **16**. As a person skilled in the art would appreciate, an elevator car would typically have one guidance assembly cluster **100** mounted on each side at the bottom of the elevator car, and may also have one guidance assembly cluster **100** mounted on each side at the top of the elevator car, for a total of four guidance assembly clusters, although the top of the elevator car may instead use purely passive roller guide assemblies of the prior art. The scope of the invention is not intended to be limited to an elevator system having a particular number or placement of guidance assembly clusters **100** in the elevator system. For the purpose of describing the invention, a complete description of the side-to-side active roller guide assembly **108** is provided below in detail in FIGS. **4**–**9**. With an understanding of the side-to-side active roller guide assembly **108**, any person skilled in the art would appreciate the details of the front-to-back active roller guide assembly **102**, which is not shown or described in the same detail.

FIG. **4** shows the side-to-side active roller guide assembly **108** in greater detail, and includes two end plates **120** attached to each end of a main support plate **122**. A shaft assembly **124** is supported between the end plates **120** for pivotal movement and includes a shaft **126** (see also FIG. **6**). The shaft **126** extends through both end plates **120** and is supported at both ends by end plate bearings **128** which are attached to the outside of each end plate **120**. The shaft assembly **124** also includes a pivot pin support beam **130** and a magnet iron support beam **132** which are both mounted between pivotally-mounted plates **134**, adjacent and parallel to the shaft **126**. A safety plug **136** is mounted on the end of the shaft **126** opposite to the guide wheel **110** to protect the end of the shaft **126** and prevent injury to personnel.

The side-to-side roller guide assembly **108** also includes three solenoids **138** mounted to the bottom of the main support plate **122**. The solenoids **138** are located directly beneath the pivot pin support beam **130** and are arranged in a line parallel to the shaft **126**. Lower spring seats **139** are mounted directly on top of each solenoid **138** and protrude through solenoid apertures **140** (see FIG. **6**). Compression springs **141** are seated on top of each lower spring seat **139**, perpendicular to the main support plate **122** and directly beneath the pivot pin support beam **130**. In one embodiment of the present invention, all three solenoids **138** are the same and all three compression springs **141** within the side-to-side assembly **108** have the same spring constant to provide equal spring force. In one embodiment, all three solenoids **138** may be separately actuated to produce a respective 184 Newton force, and all three compression springs **141** have a combined spring constant of 26 Newtons per millimeter. In operation, when one solenoid **138** is actuated a 184 Newton force is applied on a respective compression spring **141**; when two solenoids **138** are actuated a 184 Newton force is applied on two respective compression springs **141**, and when three solenoids **138** are actuated a 184 Newton force is applied on three respective compression springs **141**. However, any person skilled in the art would appreciate that the solenoid force or the spring constant may be changed without departing from the scope of the present invention.

FIG. **5** shows the side-to-side roller guide assembly **108** in further detail from a side view. The side-to-side roller guide assembly **108** also includes a lever **142** having an upper end **142**A and a lower end **142**B (see also FIG. **6**). The upper end **142**A is attached to one end of the shaft **126**, while the lower end **142**B supports the guide wheel **110**. The lever **142** converts rotary motion of the shaft assembly **124** into translational movement of the guide wheel **110** so that the guide wheel **110** may be held in contact with the guide rail (see FIG. **1**). The side-to-side roller guide assembly **108** also includes a hard stop **144** and a hard stop pin **145** to prevent over-rotation of the shaft assembly **124**. (See also FIG. **9**.) The hard stop **144** includes conical spring washers **146** located between a contact surface **147** and the hard stop **144** for cushioning any contact between the hard stop pin **145** and the contact surface **147**.

A magnet iron **148** is mounted to the magnet iron support beam **132** (FIG. **4**) directly above an electromagnet **150** which is mounted to the main support plate **122**, for providing an attractive magnetic force with respect to the magnetic iron **148**, as best seen in FIG. **6**.

FIG. **6** is an exploded view of the side-to-side assembly **108**. Each of the end plates **120** includes support grooves **120**A for supporting the main support plate **122**. In addition, the main support plate **122** includes a hard stop aperture **151** into which the hard stop **144** is mounted. The shaft **126** has two shaft ends **126**A, **126**B. Each shaft end **126**A, **126**B is tapered and dimensioned to fit within a nonlocking tapered bore **152** in lever **142**. Each shaft end **126**B includes a key slot (unlabelled) for receiving an alignment key **126**B', provided to align the lever **142** with the shaft **126**. A safety plug **136** is mounted to the shaft end **126**A. In addition, the side-to-side roller assembly **108** also includes a junction box assembly **154** for providing the necessary electrical connections for the side-to-side roller guide assembly **108**.

The side-to-side roller guide assembly **108** also includes a main cover **156** and a side cover **158** for protecting it from undesirable elements, like dirt and dust. The side cover **158** includes an access aperture **158**A which is covered by an access panel **160** for accessing the junction box assembly **154**.

FIG. **7** shows a view of the solenoid assembly and the electromagnet assembly. As shown, each solenoid **138** includes a solenoid shaft **162** which steps up and down vertically as the solenoid **138** is energized and de-energized. Each solenoid **138** has a lower spring seat **139** coupled to the solenoid shaft **162** that supports and retains the compression spring **141**. An upper spring seat **164** is mounted on top of each compression spring **141** and includes a flange **164**A for retaining the compression spring **141**.

The pivot pin support beam **130** includes a cavity **166**. A spherical pivot pin **168** is mounted through the pivot pin support beam **130** and held in contact with a spherical socket **170** formed in the upper spring seat **164**. The cavity **166** is shaped and dimensioned so that the pivot pin support beam **130** overlaps the upper spring seat **164** and holds it in place as the shaft assembly **124** rotates. This arrangement maintains the pivot pin support beam's **130** smooth continuous contact with the upper spring seat **164** as the shaft assembly **124** rotates.

As shown, the electromagnet **150** is mounted directly to the main support plate **122**. The magnet iron **148** is mounted to the magnet iron support beam **132** which in turn is connected between the pivotally-mounted plates **134**. The magnet iron **148** includes an angled bottom surface **148**A which becomes aligned parallel to the top of the electromagnet **150** as the shaft assembly **124** pivots. A gap sensor (not shown) is mounted in a cavity within a printed circuit board **180** that is mounted directly on top of the electromagnet **150**. The gap sensor is arranged in an airgap between the electromagnet **150** and the magnet iron **148** for sensing the flux density in the airgap between the electromagnet **150** and the magnet iron **148** to provide a sensed elevator car position signal to the elevator car controller **20** in FIG. **1**. The magnet iron **148** is typically made of iron, and the magnet iron support beam **132** is typically made from a non-magnetic attracting material such as aluminum, although the scope of the invention is not intended to be limited to any particular materials. Moreover, the scope of the invention is not intended to be limited to only an embodiment having compression springs **141** and attractive electromagnets **150**. For example, as any person skilled in the art would appreciate, if the positions of the solenoid actuator and the electromagnet were reversed with respect to the side-to-side guide wheel **110**, then expansion springs in combination with repulsive electromagnets could be used.

FIGS. **8–9** show how the side-to-side roller guide assembly is calibrated. As shown, a calibration pin **174** is inserted into a calibration pin slot **178** provided at the ends of the pivotally-mounted plate **134**. The calibration pin **174** locks the shaft assembly **124** into a horizontal position so that it cannot move. In this position, a calibration wedge **182** is inserted between the electromagnet **150** and the magnet iron **148** so that magnet shims **184** (see FIG. **6**) can be inserted to set the proper location of the magnet iron **148**. The setting of the magnet iron **148** position is typically done during the manufacturing and assembly process at the manufacturer but may of course be done at any other time that it is necessary.

In FIG. **9**, the calibration pin **174** also performs the function of setting the hard stop gap. For example, once the calibration pin **174** has been properly installed, the shaft assembly **124** cannot pivot. Accordingly, a hard stop gap distance indicated by the letter "G" can be set by adjusting the depth of the hard stop pin **145**.

FIG. **10** shows in detail the elevator car controller **20** (FIG. **1**) as an ARG control board hardware block diagram that provides centering and vibration control with respect to at least one axis of the elevator car **12**. Typically, an elevator car is controlled with respect to one or more axes, including side-to-side and/or front-to-back control. Elevator car controllers for controlling an elevator car about one or more axes are known in the art, and may include controllers shown and described in the aforementioned United States patents, as well as U.S. Pat. Nos. 5,308,938; 5,329,077; 5,367,132; 5,373,123; 5,400,872; as well as United States patent applications presently pending in the Patent Office, including Ser. Nos. 08/292,660, filed Aug. 18, 1994, and 08/688,918, filed Jul. 31, 1996, all hereby also incorporated by reference. Moreover, ARG control boards for respective axes may be coupled together to share information via an ARG axis interface control circuit. The scope of the invention is not intended to be limited to any particular axis, or any particular number of axes of the elevator car control.

In FIG. **10**, the ARG control board hardware block diagram is a microprocessor-based system having a 80186 processor that provides software control. FIG. **11** is a block diagram of a software algorithm for providing centering control of the elevator **12** in FIG. **1**. FIG. **12**(A) shows a vibration control overview which may be used in an elevator car controller **20** in FIGS. **1–2**, and FIG. **12**(B) shows a flux sensing overview which may be used to provide filter_flux force signals **1** and **2** in FIG. **12**(A). In FIGS. **12**(A) and **12**(B), the software blocks are generally indicated by dotted lines. The 80186 processor in FIG. **10** performs the software algorithm in FIGS. **11**, **12**(A) and **12**(B). Any person skilled in the art would appreciate that many of the blocks shown in FIGS. **10**, **11**, **12**(A) and **12**(B) may be implemented as either hardware or software. The scope of the invention is not intended to be limited to any particular manner in which an elevator control function is implemented, i.e. by either hardware or software. FIGS. **10**, **11**, **12**(A) and **12**(B) are described below in conjunction with one another.

As shown in FIG. **11**, a CALCULATE MAGNET #1 GAP block responds to a Flux_Force_**1** signal (see FIG. **12**(B) and a Magnet_Current_**1** signal (see FIG. **10**), for providing a Vib_Mag_Gap_**1** signal. Similarly, a CALCULATE MAGNET #2 GAP block responds to a Flux_Force_**2** signal and a Magnet_Current_**2** signal, for providing a Vib_Mag_Gap_**2** signal. A CALCULATE GAP AVERAGE block responds to the Vib_Mag_Gap_**1** signal and the Vib_Mag_Gap_**2** signal, for providing a Vib_Mag_Gap_Avg signal. A CALCULATE GAP DIFFERENCE block responds to the Vib_Mag_Gap_1 signal and the Vib_Mag_Gap_2 signal, for providing a Gap_Diff signal. The CONDITIONING CIRCUIT and CONDITIONING MAGNET #1, #2 blocks in FIG. **10** provide front end signal conditioning for the CALCULATE MAGNET #1 GAP block, the CALCULATE MAGNET #2 GAP block, the CALCULATE GAP AVERAGE block and the CALCULATE GAP DIFFERENCE block in FIG. **11**.

The Vib_Mag_Gap_Avg signal and the Gap_Dif signal are processed by the 80186 processor in FIG. **10**. As shown in FIG. **11**, a software (S/W) Low Pass block transforms the Vib_Mag_Gap_Avg signal into a Filter_Vib_Mag_Gap_Avg signal, and a second S/W Low Pass block transforms the Gap_Diff signal into a Filter_Pos_Error signal. A CALCULATE DBG DELTA software block transforms the Filter_Vib_Mag_Gap_Avg signal into a DBG_Delta signal. (Note that the acronym "DBG" means "Distance Between Guides".) A CORRECTED POSITION ERROR software block transforms the Filter_Pos_Error signal into a Corrected_Position_Error signal. A CALCULATE DELTA FORCE software block transforms the DBG_Delta signal and the Corrected_Position_Error signal into a Delta_Force signal. An UPDATE CENTERING FORCE COMMAND software block transforms the Delta_Force signal and an Old_Centering_Force signal into a Centering_Force signal. A CONVERT FORCE for Magnet/Solenoid software block transforms the Centering_Force signal into a Solenoid_Logical CMD signal and a Magnet-Force signal. The Solenoid_Logical CMD signal is communicated via the BUS in FIG. **10** to the solenoid PWM hardware. (Note that "CMD" means "command".) The Solenoid_Logic CMD signal contains information about whether one or more solenoids will be activated or deactivated for centering control to adjust the position of the elevator car in relation to the guide rails, and the Magnet-Force signal contains information about whether the one or more electromagnets will be activated or deactivated for centering the car between each roller and a respective guide rail. A CONVERT LOGICAL TO PWM COMMANDS block in FIG. **11** corresponds to the PWM Logic block in FIG. **10**. The CONVERT LOGICAL TO PWM COMMANDS block in FIG. **11** responds to the Solenoid_Logical CMD signal, for providing a SOL_**1**_PWM signal, a SOL_**2**_PWM signal, and a SOL_**3**_PWM signal, which are further processed by the GATE DRIVER and POWER SWITCH blocks in FIG. **10** for providing a solenoid control signal to solenoid banks A and B. A S/W Low Pass filter block transforms the Magnet-Force signal into an Offset_Force signal that is provided to a software summing junction shown and described with respect to FIG. **12**(A).

The Microfiche Appendix includes a computer program listing of a centering control module to perform the software algorithm in FIG. **11**. The program listing in the Microfiche Appendix is an algorithm for apportioning solenoid and magnet force for centering control. In general, in the computer program listing, a total centering force is computed as a function of a delta force, and a centering force is clamped at a maximum centering force if necessary. A check is made to determine whether or not a maximum number of solenoids should be turned on. A variable PerSolenoidForce/2 is determined which represents a range where only the electromagnets are used for centering control. For example, if the PerSolenoidForce equals 184 Newtons, then the PerSolenoidForce/2 equals 92 N. If the centering force plus the PerSolenoidForce/2 is greater than or equal to the maximum centering force, then the number of solenoids turned on equals the number of solenoids. Otherwise, the number of solenoids turned on is calculated directly from the total centering force required together with the solenoid force per solenoid using the equation:

$$\#OfSolenoidsOn = \frac{(CenteringForce + (PerSolenoidForce/2)}{(PerSolenoidForce)}.$$

Next, the amount of force associated with the number of solenoids turned on is calculated using the equation:

SolenoidForce=#OfSolenoidsOn*PerSolenoidForce;

Finally, a magnet force required to take up the balance of the total centering force is calculated using the relationship:

MagnetForce=CenteringForce−SolenoidForce;

The above equations are valid for the case where the centering force required is positive. For the case where centering force is negative, the equations are the same, except that the sign of the variables CenteringForce and SolenoidForce is negated. These equations are consistent with that shown and described with respect to the centering force diagram shown in FIG. **13**, discussed below.

FIG. **12**(A) shows a vibration control overview which may be used in the elevator car controller **20** in FIGS. **1–2**. In such an embodiment, a hybrid approach is used in which the electromagnet formerly used for vibration control only also supplies some of the centering force required, i.e. fine position control. For example, in the side-to-side operation, each solenoid **138** (see FIG. **6**) only coarsely provides the same single force. The fine resolution of total centering force is provided by the electromagnets **150** (see also **34**, **36** in FIG. **1**) on each side of the elevator car. This fine resolution force is commanded by an "Offset Force" command signal from the S/W Low Pass block of FIG. **11** and shown entering a software summing junction (Σ) described below as the "Offset Force" (From Centering Control) in FIG. **12**(A), and used in conjunction with the solenoid force for centering the car.

In particular, FIGS. **12**(A) and (B) show an overview of a side-to-side or a front-to-back hybrid vibration/fine-centering control system for the elevator car. In general, the vibration/fine-centering control system is a feedback control loop in which an acceleration of the elevator car **12** (FIG. **1**) and both a flux and current of the electromagnet **150** (FIG. **6**) are sensed and used to determine the airgap and a counteracting force to negate the acceleration on the elevator car **12** and, in conjunction with the coarse solenoid actuators, to keep the car finely centered in the hoistway using fine position resolution control. As shown, the vibration control system is implemented in both hardware and software, with the solid boxes being representative of a hardware implementation and dotted boxes being representative of a software implementation. The software blocks are performed by the 80186 processor shown in FIG. **10**. The scope of the invention is not intended to be limited in any way to whether a particular function is implemented in hardware or software.

Any person skilled in the elevator active suspension control art would appreciate after reviewing the vibration control system shown in FIG. **12**(A) how an elevator car acceleration is sensed and converted into the counteracting force to negate the acceleration on the elevator car **12**. In view of this, only a brief description of the vibration control system shown in FIG. **12**(A) is provided herein. The vibration control system includes a VT1/SCA **11** block that represents a sensor for sensing an actual acceleration of the elevator car, for providing a signal representing an actual elevator car acceleration. An OFFSET AMP block is a signal processing block that shifts the actual elevator car acceleration signal, amplifies it to usable voltage level, and provides an amplified signal corresponding to the actual elevator car acceleration. In effect, the shift is to a full scale range because the output of the accelerometer is in microvolts, so the second block is a signal processing block to provide a desirable voltage range. A LOW PASS FILTER block filters out noise and irrelevant signals beyond 100 Hertz, for providing a filtered analog signal. The CONDITIONING CIRCUIT block in FIG. **10** includes the VT1/SCA **11** block, the OFFSET AMP block and the LOW PASS FILTER block in FIG. **12**(A).

A MUX CHANNEL (A/D) block multiplexes and performs an analog to digital conversion, and provides a digital filtered acceleration signal corresponding to the actual elevator car acceleration to a microprocessor for processing the signal, as shown by the dotted blocks. The 16 CHANNEL MUX in FIG. **10** includes the MUX CHANNEL (A/D) block in FIG. **12**(A).

The 80186 processor in FIG. **10** executes a COMP_ACCEL OFFSET COMPENSATION software block for providing a Comp_Accel signal to an ADJUSTABLE BAND PASS FILTER block, which processes the digital filtered acceleration signal, and provides a Filtered_Comp_Accel signal to an ACCEL TO FORCE CONVERSION software block for performing the acceleration to force conversion and providing an Accel_Force signal. A VIBRATION GAIN software block processes the Accel_Force signal to adjust the gain in the control loop, and provides a Scaled_Accel_Force signal to a VIBRATION REGULATOR software block. The VIBRATION REGULATOR software block processes the Scaled_Accel_Force signal and provides an Accel_Force_Command signal to a software summing junction in another software block. In effect, the Accel_Force_Command signal is a signal representing a force that was converted from an accelerometer input signal, and is commonly known as a force dictation signal, i.e. it represents the amount of force needed to counteract the sensed acceleration.

The SUMMING JUNCTION software processes four signals, the Accel_Force_Command signal, a Filter_Flux_Force_**1** signal from a filtered_flux_force_**1** circuit shown and described with respect to FIG. **12**(B), a Filter_Flux_Force_**2** signal from a filtered_flux_force_**2** circuit also shown and described with respect to FIG. **12**(B), and the OFFSET_FORCE signal from an elevator car centering side-to-side control, discussed above. The SUMMING JUNCTION software responds to these four signals and provides a Force_Error signal to a FORCE GAIN & PI REGULATOR block. The FORCE GAIN & PI REGULATOR software block processes the Force_Error signal, and provides a force gain and proportional/integral regulator signal to a FORCE COMMAND AND CMD LIMIT LOGIC PWM LOGIC block.

The magnet current feedback loop of the vibration control also includes the following: A MAGNET #1: CURRENT/VOLTAGE & LPF block that responds to an IMAG**1** current from electromagnet #1 and provides a current/voltage converted and low pass filtered signal. A MUX CHANNEL:**3** (A/D) block multiplexes and A-to-D converts the current/

voltage converted and low pass filtered signal and provides a digital filtered IMAG1 signal to the microprocessor for processing. The digital filtered IMAG1 signal represents a feedback current of the current being provided to electromagnet #1. Similarly, a MAGNET #2: CURRENT/VOLTAGE & LPF block responds to an IMAG2 current from electromagnet #2 and provides a current/voltage converted and low pass filtered signal. A MUX CHANNEL (A/D) block multiplexes and A-to-D converts the current/voltage converted and low pass filtered signal and provides a digital filtered IMAG2 signal to the microprocessor for processing. Similarly, the digital filtered IMAG2 signal represents a feedback current of the current being provided to electromagnet #2. The conditioning MAGNET #1 AND #2 BLOCKS FIG. **10** correspond to the MAGNET #1: CURRENT/VOLTAGE & LPF and MAGNET #2: CURRENT/VOLTAGE & LPF blocks, and the 16 CHANNEL MUX in FIG. **10** includes the MUX CHANNEL:**3** and the MUX CHANNEL:**4** (A/D) blocks.

The FORCE COMMAND & CMD LIMIT PWM LOGIC software block responds to the Force Gain & Proportional/Integral Regulator signal, the digital filtered IMAG1 signal, the digital filtered IMAG2 signal and the sensed 150 voltage signal from the 150V Sense Circuit of FIG. **10**, for providing a Magnet_command_**1** signal and a Magnet_command_**2** signal. A MAGNET DRIVER #1 AND #2 software block responds to the Magnet_command_**1** signal and the Magnet_command_**2** signal, for providing a PWM magnet driver signal for an electromagnet #1 and a PWM magnet driver signal for an electromagnet #2 to generate the counteracting force to negate the acceleration on the elevator car **12** and for fine centering. The electromagnet #1 may be an electromagnet such as the electromagnet **34** of FIG. **1** and the electromagnet #2 may be an electromagnet such as the electromagnet **36**, as also shown in FIG. **1**. The electromagnets may also be electromagnets on each side of a rail for front-to-back guide. The magnet driver signals for electromagnets #1 and #2 are further processed by GATE DRIVER blocks and POWER SWITCH blocks associated with electromagnets #1 and #2 in FIG. **10**, for providing gate driver and power switch signals to electromagnets #1, #2 for providing fine centering and vibration control of the elevator car **12**.

FIG. **12**(B) represents circuits for force feedback for electromagnet #1 and electromagnet #2, such as electromagnets **34**, **36**, or vice versa in FIG. **1**. The circuit for electromagnet #1 responds to actual sensed flux from electromagnet #1 and provides a Filter_Flux_Force_**1** signal to the summing junction software of FIG. **12**(A). Similarly, the circuit for electromagnet #2 responds to actual sensed flux from electromagnet #2 and provides a Filter_Flux_Force_**2** signal to the summing junction software. In both circuits, the circuit BELL GH-600 represents a flux sensor such as **180** shown in FIG. **6**, which converts flux to voltage, and provides a sensed flux signal. The sensed flux signal is processed in a signal processing stage, a gain stage and a roll-off and a filtering stage for providing a voltage signal in a useful range like 0–10 volts. The voltage signal is then multiplexed and converted to a digital 12-bit signal, then provided to the microprocessor for software processing, where it is calibrated, squared, converted from a flux signal to a force signal, filtered, and provided as the Filtered_Flux_Force signal_l, discussed above. The CONDITIONING CIRCUIT block in FIG. **10** includes the BELL GH-600 and DIFF AMP: GAIN blocks in FIG. **12**(B). The 16 CHANNEL MUX in FIG. **10** includes the MUX CHANNEL (A/D) blocks in FIG. **12**(B). The circuit for electromagnet #2 operates in a similar manner.

This approach of using a plurality of identically sized solenoids for coarse position control in conjunction with fine position control using electromagnets has a number of benefits over alternative embodiments using a plurality of coarse and finely graded solenoids, discussed below in connection with FIGS. **14–18**, for the following reasons:

fewer solenoids

less maximum solenoid force required

all solenoids, springs and spring seats identical

better force resolution

softer effective spring constants possible at the roller (i.e. more practical implementation, especially side-to-side)

more graceful degradation in the event of a bad solenoid.

As shown, the control system has an acceleration loop and a force loop.

FIG. **13** is a graph depicting centering output force(s) on the ordinate versus centering force required on the abscissa and is presented in this way in order to show how a composite of coarse solenoid and fine electromagnet forces result in a smooth "effective force". In general, the solid diagonal line represents the sum of the coarse (step) solenoid and the fine (ramp) electromagnet forces. The staircase curve superimposed on the solid diagonal line represents step changes in solenoid force (184 Newtons) each time a solenoid is separately activated (or deactivated). The series of diagonal lines (sawtooth) superimposed on the abscissa represents the force output of the electromagnets shifted as a function of the level of centering force required of the solenoids by the control of FIG. **11**. The notations **(1)** and **(2)** indicate which electromagnet is activated, i.e., depending on whether the effective force exerted is positive or negative, respectively. Electromagnet **(2)** is used to apply force which is referred to as a "negative" force for the purpose of establishing a convention.

In particular, the graph in FIG. **13** shows a side-to-side centering control for an elevator system using both the solenoids and the electromagnets such as those shown and described above. FIG. **13** has a total centering output in the ordinate or vertical axis as a function of a required centering force on the horizontal X axis. The top right or first quadrant of the graph represents the actuation of one or more right side solenoids, and the lower left quadrant of the graph represents the actuation of one or more left side solenoids. (In a front-to-back centering control, the top right quadrant of the graph may represent the actuation of one or more back side solenoids, and the lower left quadrant of the graph represents the actuation of one or more front side solenoids.) The force of the left and right electromagnets is represented as a sawtooth function of lines labelled "1" and "2" respectively in the center and along the X axis or abscissa of the graph. The diagonal line from the lower left or third quadrant to the top right or first quadrant in FIG. **13** represents an effective force, which is determined by summing together the solenoid force and the force from the electromagnets. The solenoid force has a step function characteristic having steps of 184 Newtons. An arrow labeled "solenoid force" points to a solenoid step change in solenoid force. As shown, an arrow labeled "vibration electromagnets **(1)**; **(2)**" points to the sawtooth function in the middle of the graph representing the forces of the electromagnets within a similar range of −92 Newtons<0<+92 Newtons on the ordinate.

The cooperation between the solenoids and the magnets will be briefly described. In the vicinity of the origin, the electromagnets **(1)** and **(2)** work in a range of −92 Newtons to +92 Newtons, and no solenoids are turned on the left side or the right side of the elevator car. As one example, the electromagnet E(**1**) may be supplying the centering force in a range of 0 to +92 Newtons and therefore be operating in the initial ramp portion of the effective force line in quadrant **1**, before any solenoid is actuated. At 92 Newtons there is a dotted line showing one solenoid activated on the right side of the elevator car to provide a 184 Newtons change in centering force. At the same time, electromagnet (**1**) is deactuated at 92 Newtons and electromagnet (**2**) is activated on the left side of the elevator car at 92 Newtons, so there is no effective change in force with respect to the elevator car. If more or less centering force is needed, then electromagnet (**2**) may be respectively decreased or increased within a range of 0–92 Newtons, i.e., back to the origin, or further along the effective force line into quadrant **1**. Eventually when the force exceeds the range of 0–92 Newtons, then the solenoid E(**1**) has to be turned on or off or E(**1**) is reactivated and the appropriate electromagnets (**1**) or (**2**) will be actuated. As will be evident to any person of skill in the art, the same process can be continued in either direction in stepwise fashion and by actuating and deactuating the appropriate actuators.

Parallel Binomial Solenoids

FIG. **14** shows an embodiment of the present invention having solenoids with different sizes arranged in a parallel connection. The individual solenoids are all connected to a main supporting plate, which provides good heat transfer characteristics for the assembly. Each solenoid pushes on the reacting arm through its own linear spring. The embodiment is similar to the embodiment shown and described with respect to FIGS. **3**–**13** above in that the parallel solenoids are arranged in a line.

The selection of solenoid size, working stroke, and individual spring constant provides the ability to command $2^{(n-1)}$ different force levels through the actuator, as a function of an "n" bit command, where each command bit corresponds to one of the solenoids. The "weighting" of each solenoid/spring unit can be made to be binary. For example, in the case of four bits, the largest solenoid commands 8⁄15 of the total force, the next largest solenoid commands 4⁄15 of the total force, etc.

In the case of the Active Roller Guide, the solenoids and Digital Linear Magnetic Actuators are commanded by a controller which calculates the gap width in the vibration control magnets by sensing the flux in the vibration magnet gap, and comparing this with the current in the vibration magnet winding (vibration magnets not shown in FIG. **14**). These gaps are sampled many times a second, and the average gap is calculated by the controller for the purpose of determining whether or not the car is centered relative to the rails.

When power is applied to a solenoid, it extends by the amount of its working stroke. In general, the product of the solenoid stroke and its associated spring constant must represent a doubling of force with each additional solenoid in the assembly, in order to maintain a binary weighting. The resolution of the solenoid configuration is defined by the smallest solenoid/spring combination, and the dynamic range of the solenoid configuration is defined by the number of solenoids, in conjunction with the resolution.

In equation form, the solenoid configuration force F_{so1} generated by the parallel DLMA can be represented by the following:

$$F_{sol} = \sum_{a=1}^{a=n} Ca^{*}Xa^{*}Ka$$

where n is the number of solenoids, Ca is the binary command to solenoid n, Xa is the stroke of solenoid n, and Ka is the spring constant of the spring associated with a.

This series of actuator strokes as represented by a binary number, (0 to 16 in FIG. **14**) makes the command of the actuator very simple to implement with standard digital design approaches.

The worst case force transmitted through the largest solenoid is only 8⁄15 of the total. This represents a considerable relaxation of force requirements on individual solenoids in the actuator, as compared with the series stack of solenoids, discussed below.

In an ordinary magnetic actuator, supplying static holding forces across the normal working gap of the magnet (assuming that the spring constant is the same as the effective spring constant at the roller guide) would lead to very large magnet designs, (at 1:1 mechanical advantage) which would be expensive in terms of weight and power consumption. In the case of the Digital Linear Magnetic Actuator, however, each solenoid is operated at minimum gap, resulting in lower power consumption and smaller size.

FIG. **15** illustrates a command and control technique for the embodiment shown in FIG. **14** that may be implemented by an elevator car controller such as **20** shown in FIGS. **1**–**2**. The supply voltage for the solenoids is a Pulse Width Modulated rectangular waveform. When any given combination of solenoids is fully engaged, the reluctance of each activated solenoid is at maximum, and the DC current demand for a given PWM duty cycle is at minimum. If the stroke of the actuator needs to be changed, then the PWM duty cycle is reset, and the new digital command is selected. The appropriate actuators engage and disengage, and then in the next update cycle, the PWM duty cycle is set to a new value based upon the following:

PWM Duty Cycle=A*####,

where A is a constant which is a function of solenoid vendor holding voltage, and where #### is the binary stroke commanded to the actuator, corresponding to static holding force.

FIG. **16** provides an illustration of the PWM control sequence for the embodiment shown in FIG. **14** with each actuator command update. First, the position reference is checked to determine if more or less centering force is required. If more is required, then the current actuator command is checked to determine if the command is at maximum (1111). If so, then no more force can be applied. If the command is not at maximum, then the PWM command is set to 100%, and the actuator command count is incremented by one.

In the case of less centering force being required, the command count is checked to confirm that it is greater than minimum (0000). If so, then the PWM command is set to 100% and the actuator command is decremented by one. If the command is already 0000, then no less force can be applied on this roller; centering force must be applied against the opposite roller.

After making the adjustment to the actuator command count, after a delay period, the Pulse Width Modulated command is set to A times the actuator command count, for the purpose of minimum power dissipation.

When the Digital Linear Magnetic Actuator is commanded in this manner, the net change is 1⁄15 of maximum per update. For the Active Roller Guide application, this granularity of centering control is adequate. The step change in the force through the spring can easily be compensated for by the vibration control magnets, in conjunction with acceleration feedback from the elevator car frame.

In summary, the parallel Digital Linear Magnetic Actuator provides the benefits of a magnetic actuation device which is reliable, while circumventing the power consumption problem, by operating individual devices at minimum gap. Minimum power is required for keeping the magnets from "ungluing" during dynamic events by commanding the Pulse Width Modulated supply as a function of the total static force transmitted through the actuator, plus margin. The heat transfer characteristics are favorable, and alignment problems are avoided. The maximum force requirements on individual solenoids are reduced.

The implementation described here uses linear springs; the concept is equally applicable when applied to spiral springs and rotary solenoids, although this approach would require multiple mounting plates.

Series Binomial Stack of Solenoids

FIG. **17** shows a cut-away view of an implementation of a Digital Linear Magnetic Actuator assembly, having DLMAs in series with the suspension spring. The DLMAs are commanded by a controller which senses the gap width in the vibration control magnets by sensing the flux in the vibration magnet gap, and comparing this with the current in the vibration magnet winding. These gaps are sampled many times a second, and the average gap is calculated by the controller for the purpose of determining whether or not the car is centered relative to the rails. The position sampling technique is the same as the one described above.

As shown, the implementation is comprised of a linear coil spring with a spring constant K_s. A cup is mounted inside the coil spring which serves as a receptacle for multiple solenoids. In the example shown, four are used. The solenoids are stacked "in series" inside the receptacle, and each solenoid has a different stroke as summarized below:

Examples of DLMA solenoid strokes are:

S4.0: 4.0 mm

S2.0: 2.0 mm

S1.0: 1.0 mm

S0.5: 0.5 mm

When power is applied to a solenoid, it extends by the amount associated with its label, i.e. 4, 2, 1, or 0.5 mm. This series is a specific example; in general, to maintain binary weighting, the series of solenoid strokes must represent a doubling of strokes with each additional solenoid. The resolution of the solenoid stack is defined by the smallest stroke solenoid, and the dynamic range of the solenoid stack is defined by the number of solenoids, in conjunction with the resolution.

Thus, for this example, the actuator provides a 0 to 7.5 mm stroke, with 0.5 mm resolution, as summarized by the table below:

S4	S2	S1	S0.5	Stroke	Force
0	0	0	0	0	0
0	0	0	1	0.5	Ks*Stroke
0	0	1	0	1.0	Ks*Stroke
0	0	1	1	1.5	Ks*Stroke
0	1	0	0	2.0	Ks*Stroke
0	1	0	1	2.5	Ks*Stroke

-continued

S4	S2	S1	S0.5	Stroke	Force
0	1	1	0	3.0	Ks*Stroke
0	1	1	1	3.5	Ks*Stroke
1	0	0	0	4.0	Ks*Stroke
1	0	0	1	4.5	Ks*Stroke
1	0	1	0	5.0	Ks*Stroke
1	0	1	1	5.5	Ks*Stroke
1	1	0	0	6.0	Ks*Stroke
1	1	0	1	6.5	Ks*Stroke
1	1	1	0	7.0	Ks*Stroke
1	1	1	1	7.5	ks*Stroke

This series of actuator strokes can be represented by a 4 bit binary number, 0 to 16, and makes the command of the actuator very simple to implement with standard digital design approaches.

Since all of the solenoids are in series with the spring, the entire force transmitted through the spring must be carried by each solenoid. For the ARG application, the imbalance force at the rail/roller interface with 1000 lbs. of imbalance load is approximately 145 lbs. To this static imbalance force, about 12% must be added to handle worst case acceleration profiles, plus the worst case dynamic force provided by the vibration magnet, which corresponds to the worst case additional spring compression with worst case side-to-side rail deviation. For the ARG application, this total force amounts to approximately 210 lbs.

In an ordinary magnetic actuator, supplying these static holding forces across a gap of this magnitude (assuming that the spring constant is the same as the effective spring constant at the roller guide) would lead to very large magnet designs, which would be expensive in terms of weight and power consumption. In the case of the DLMA, however, each solenoid is operated at minimum gap, resulting in lower power consumption and smaller size.

FIG. **18** illustrates a command and control technique for the embodiment shown in FIG. **17** that may be implemented by an elevator car controller such as **20** shown in FIGS. **1–2**. The supply voltage for the solenoids is a PWM rectangular waveform. When any given combination of solenoids is fully engaged, the reluctance of each activated solenoid is at maximum, and the DC current demand for a given PWM duty cycle is at minimum. If the stroke of the actuator needs to be changed, then the PWM duty cycle is set to 100%, and the new digital command is selected. The appropriate actuators engage and disengage, and then in the next update cycle, the PWM duty cycle is set to a new value based upon the following:

$$\text{PWM Duty Cycle} = A * \#\#\#\#,$$

where A is a constant which is a function of solenoid vendor holding voltage, and where #### is the binary stroke commanded to the actuator, corresponding to static holding force.

The solenoid logic command determination for the stacked series of DLMAs is substantially the same as that for the parallel DLMAs shown in FIG. **16**. For example, first, the position reference is checked to determine if more or less centering force is required. If more is required, then the current actuator command is checked to determine if the command is at maximum (1111). If so, then no more force can be applied. If the command is not at maximum, then the PWM command is set to 100%, and the actuator command count is incremented by one.

In the case of less centering force required, the command count is checked to confirm that it is greater than minimum

(0000). If so, then the PWM command is set to 100% and the actuator command is decremented by one. If the command is already 0000, then no less force can be applied on this roller; the centering force must be applied against the opposite roller.

After making the adjustment to the actuator command count, after a delay period, the Pulse Width Modulated command is set to A times the actuator command count, for the purpose of minimum power dissipation.

When the DLMA is commanded in this manner, the net change in spring length is 0.5 mm per update. For the ARG application, this granularity of centering control is adequate. The step change in the force through the spring can easily be compensated for by the vibration control magnets, in conjunction with acceleration feedback from the elevator car frame.

In summary, the stacked series of DLMAs provides the benefits of a magnetic actuation device, while circumventing the power consumption problem, by operating individual devices at minimum gap. Minimum power is required for keeping the magnets from "ungluing" during dynamic events by commanding the PWM supply as a function of the total static force transmitted through the actuator, plus margin.

The implementation described here uses linear springs, the concept is equally applicable when applied to spiral springs and rotary solenoids.

Some drawbacks to the stacked series approach include the fact that there are mounting problems associated with keeping the stack aligned properly, and getting the heat dissipated by the solenoids out of the assembly. In addition, each solenoid must carry the full force transmitted by the stack, which drives up the size of all the solenoids. Finally, the total amount of force delivered by each solenoid is limited due to series preloading effects.

Since all of the solenoids are in series with its spring, the entire force transmitted through each spring must be carried by each solenoid. For the Active Roller Guide application, the imbalance force at the rail/roller interface with 1000 lbs. of imbalance load is approximately 145 lbs. To this static imbalance force, about 12% must be added to handle worst case acceleration profiles, plus the worst case dynamic force provided by the electromagnet for vibration control, which corresponds to the worst case additional spring compression with worst side-to-side rail deviation. For the Active Roller Guide application, this total force amounts to approximately 210 lbs.

Although the present invention has been described and discussed herein with respect to one or more embodiments, other arrangements or configurations are possible which do not depart from the spirit and scope hereof. Hence, the present invention is deemed limited only by the appended claims and the reasonable interpretation thereof.

What is claimed is:

1. A roller guide assembly for controlling the position of an elevator car in relation to guide rails of an elevator hoistway in an elevator system, comprising:

at least one solenoid means, responsive to a solenoid centering control signal from an elevator car controller containing information about a combined solenoid and electromagnet centering control force, for providing at least one solenoid centering control force; and

at least one electromagnet means, responsive to an electromagnet centering and vibration control signal from the elevator car controller containing information about the combined solenoid and electromagnet centering control force, for providing at least one electromagnet centering and vibration control force;

wherein the combined solenoid and electromagnet centering control force to position the elevator car in relation to the guide rails is a counterbalancing force including a combination of the solenoid centering control force and the electromagnet centering and vibration control force.

2. A roller guide assembly according to claim **1**, wherein said at least one solenoid means further comprises at least one coil spring that responds to the at least one solenoid centering control force, for providing at least one coil spring force for adjusting the position of the elevator car in relation to the guide rails of the elevator hoistway.

3. A roller guide assembly according to claim **1**, wherein said at least one electromagnet means provides said at least one electromagnet centering and vibration control force to control the vibration between the roller of the elevator car and a guide rail of the elevator hoistway during the control of the position of the elevator car in relation to the guide rails of the elevator hoistway.

4. A roller guide assembly according to claim **1**, wherein the control of the position of the elevator car in relation to the guide rails of the elevator hoistway includes side-to-side and front-to-back centering control.

5. A roller guide assembly according to claim **1**, wherein the solenoid centering control signal and the electromagnet centering and vibration control signal includes information about the position and/or acceleration of the elevator car in relation to the guide rails of the elevator hoistway.

6. A roller guide assembly according to claim **1**,

wherein said at least one electromagnet centering and vibration control force provides a fine adjustment of the position of the elevator car in relation to the guide rails of the elevator hoistway; and

wherein said at least one solenoid centering control force provides a coarse adjustment of the position of the elevator car in relation to the guide rails of the elevator hoistway.

7. A roller guide assembly according to claim **1**, wherein said at least one solenoid means further comprises a plurality of solenoids, each for separately providing a respective solenoid force for adjusting the position of the elevator car in relation to the guide rails of the elevator hoistway.

8. A roller guide assembly according to claim **7**, wherein each of the plurality of solenoids separately provides a respective substantially equal solenoid force.

9. A roller guide assembly for an elevator system having an elevator car controller to control the position of an elevator car in relation to guide rails of the elevator hoistway and to control the vibration of a guide wheel of the elevator car and a guide rail of the elevator hoistway, comprising:

a solenoid, responsive to a solenoid centering control signal from an elevator car controller containing information about a combined solenoid and electromagnet centering control force, for providing a solenoid centering control force;

a coil spring, responsive to the solenoid centering force, for providing a coil spring centering control force to position the elevator car in relation to the guide rails of the elevator hoistway; and

at least one electromagnet, responsive to the elevator car electromagnet centering and vibration control signal from the elevator car controller containing information about the combined solenoid and electromagnet centering control force, for providing at least one electromagnet centering and vibration control force to control the centering and vibration between the guide wheel of the elevator car and the guide rail of the elevator hoistway;

wherein the combined solenoid and electromagnet centering control force to position the elevator car in relation to the guide rails is a counterbalancing force including a combination of the solenoid centering control force and the electromagnet centering and vibration control force.

10. A roller guide assembly for an elevator system having an elevator car controller to control the position of an elevator car in relation to guide rails of the elevator hoistway and to control the vibration of a guide wheel of the elevator car and a guide rail of the elevator hoistway, comprising:

a solenoid, responsive to a solenoid control signal from the elevator car controller, for providing a solenoid force;

a coil spring, responsive to the solenoid force, for providing a coil spring force to control the position of the elevator car in relation to the guide rails of the elevator hoistway; and

at least one electromagnet, responsive to the elevator car control signal, for providing at least one electromagnet force to control the vibration between the guide wheel of the elevator car and the guide rail of the elevator hoistway and to control the position of the elevator car in relation to the guide rails of the elevator hoistway;

wherein said coil spring is a linear coil spring having a spring length and a spring constant Ks, responsive to the coil spring force, for providing a variable linear coil spring force that depends on the spring length and the spring constant Ks for controlling the position of the elevator car in relation to the guide rails of the elevator hoistway.

11. A roller guide assembly according to claim **10**, wherein the control of the position of the elevator car in relation to the guide rails of the elevator hoistway includes side-to-side and front-to-back centering control.

12. A roller guide assembly according to claim **11**,

wherein said roller guide assembly further comprises a plurality of solenoids, each for separately providing a respective solenoid force for adjusting the position of the elevator car in relation to the guide rails of the elevator hoistway

wherein said roller guide assembly further comprises a plurality of coil springs, each separately responsive to the respective solenoid force, each for providing a respective coil spring force to control the position of the elevator car in relation to the guide rails of the elevator hoistway.

13. A roller guide assembly according to claim **12**, wherein each of the plurality of solenoids separately provides a respective substantially equal solenoid force.

14. A roller guide assembly according to claim **12**,

wherein the guide wheel is arranged on a pivot axis mounted on an elevator car; and

wherein the variable linear coil spring force opposes the electromagnet force about the pivot axis.

15. A roller guide assembly according to claim **10**, wherein the solenoid is a digital linear magnetic actuator.

16. A roller guide assembly according to claim **10**, wherein the roller guide assembly comprises:

a plurality of solenoids, each having a respective piston for moving with a corresponding adjustable stroke, responsive to an associated solenoid control signal from the elevator car controller, for providing a respective adjustable stroke force; and

a plurality of linear coil springs, each having a respective spring length and a respective spring constant Ks, each responsive to the respective adjustable stroke force, for providing a corresponding variable linear coil spring force that depends on the respective spring length and the respective spring constant Ks for controlling the position of the elevator car in relation to the guide rails of the elevator hoistway.

17. An active roller guide assembly according to claim **16**, wherein each pair of one of the plurality of solenoids and one of the plurality linear coil springs provides the corresponding variable linear coil spring force with substantially the same force.

18. An elevator system for controlling the position of an elevator car in relation to guide rails of an elevator hoistway, comprising:

an elevator car control, responsive to a sensed elevator car signal having information about the position of the elevator car in relation to the guide rails of the elevator hoistway, for providing a solenoid centering control signal containing information about a combined solenoid and electromagnet centering control force, and further providing an electromagnet centering and vibration control signal from the elevator car controller also containing information about the combined solenoid and electromagnet centering control force;

at least one solenoid means, responsive to the solenoid centering control signal, for providing at least one solenoid force; and

at least one electromagnet means, responsive to the electromagnet centering and vibration control signal, for providing at least one electromagnet centering and vibration control force;

wherein the combined solenoid and electromagnet centering control force to position the elevator car in relation to the guide rails includes a combination of the solenoid centering control force and the electromagnet centering and vibration control force.

19. An elevator system according to claim **18**, wherein the at least one solenoid means includes at least one solenoid connected to a corresponding coil spring.

20. An elevator system according to claim **18**, wherein the solenoid centering control signal and the electromagnet centering and vibration control signal includes information about the position and/or acceleration of the elevator car in relation to the guide rails of the elevator hoistway.

21. An elevator system according to claim **18**, wherein the solenoid centering control signal and the electromagnet centering and vibration control signal includes information about the side-to-side centering control and front-to-back centering control of the position of the elevator car in relation to the guide rails of the elevator hoistway.

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